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HLH/ATC ENGINE SHAFT SUPPORT BEARING DEVELOPMENT PROGRAM

J. W. Lenski, Jr.

Boeing Vertol Company

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EUSTIS DIRECTORATE POSITION STATEMENT

This directorate concurs with the conclusions presented herein.

The technical monitor for this effort was Mr. Wayne A. Hudgins, Heavy Lift Helicopter Project Office, Systems Support Division.

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Drive shafting which transmits power between combining transmission and engines is supported by ball bearings to structure. Because of the difficulty of providing fluid lubrication to these bearings, they have always been lubricated with grease. Present experience with the CH-46 and CH-47 helicopters has shown that these bearings will perform adequately with grease lubrication at DN operating speeds of 450,000.

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The HLH drive system will use a similar bearing configuration to support both the synchronizing shafts and the engine shafts. Preliminary designs show that the HLH shaft support bearings will be operating in the range of 650,000 DN. This is approximately 1.5 times present Boeing Vertol experience and approximately 1.2 times the maximum recorded operating speed of grease-lubricated ball bearings.

Because of the increased speed requirements for a grease-lubricated ball bearing, a test program was conducted to determine the operating characteristics, design modifications and maximum regreasing interval for a ball bearing operating at the maximum loads and speed of the HLH engine and synchronizing shaft support bearing.

This report presents the results of efforts conducted between September 1971 and June 1973 to define criteria and to design, fabricate, and test grease-lubricated bearing configurations to obtain the optimum bearing and housing design for use as the HLH engine shaft support bearing system. Tests were performed on simulated HLH engine shaft bearing housing assemblies under expected maximum loading and misalignment conditions. Technical inspection and evaluation of the test results were used for selecting the optimum bearing and housing configuration for both the HLH engine shaft support bearing and the HLH synchronizing shaft support bearing.

Testing conducted during this development program has shown that modifications to the bearing geometry and cage design, plus housing modifications for retaining the grease lubricant, will provide a bearing assembly to meet the HLH requirements. A regreasing interval time of 300 hours was achieved with these modifications.

PREFACE

The ball bearing development work presented in this report has been completed in partial fulfillment of Contract DAAJ01-71-C-0840(P40). This effort was primarily a development program to define design criteria and to design, fabricate, and test the HLH engine shaft support bearing assembly to determine the maximum regreasing interval and potential failure modes. Tests were performed on simulated bearing assemblies at design shaft speed and maximum shaft unbalance, under misalignment conditions. The test results were evaluated to determine the optimum design of the HLH engine shaft support bearing and housing assembly.

This program was conducted at Marlin-Rockwell, Division of TRW Inc., Jamestown, New York, under the technical direction of Arthur S. Irwin, Manager, Research and Development, with Harold E. Munson, Senior Research Engineer, as the principal investigator. Close contact with regard to program status and test results was maintained with Joseph W. Lenski, Senior Design Engineer, of Boeing Vertol Company. Technical direction was provided by Wayne Hudgins, Project Engineer, Eustis Directorate, U.S. Army Air Mobility Research and Development Laboratory, Fort Eustis, Virginia.

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INTRODUCTION

This bearing development* program was directed toward the development of a grease-lubricated ball bearing to operate at 632,500 DN (DN is the product of the bearing bore diameter in millimeters times the bearing speed in rpm's) in a simulated HLH engine shaft support bearing housing. The bearing chosen for testing was the MRC 111-KS deep-groove ball bearing with a 55-mm bore. The operating conditions for this test program simulated a combination of the HLH engine and synchronization shaft support applications, and were as follows:

Shaft speed	11,500 rpm (engine shaft)
Shaft misalignment	0° 15'
Shaft inclination	34° (synchronization shaft)
Shaft weight	80 pounds
Shaft unbalance	0.10 ± 0.02 inch-ounce
Lubrication	Grease
Ambient temperature	180°F
Duration	300 hours without regreasing

Prior to actual full-size bearing testing, preliminary screening tests were conducted to establish the type of grease to be used. Testing was conducted on 204 size ball bearings operated at 32,000 rpm (640,000 DN). These tests indicated that Aeroshell 22 grease, meeting Government specification MIL-G-81322, gave satisfactory performance as a lubricant under these test conditions.

To accomplish the full-scale objectives, four cage designs and two housing designs were initially proposed and put into test. As testing was performed, the need for additional bearing and housing modifications became apparent. The program was expanded until a final bearing and housing configuration was obtained to achieve a 300-hour regreasing interval. The results of these development tests are summarized in this test report.

After the completion of this program, additional testing of the final bearing and housing configuration was authorized by Contract DAAJ01-73-A-0017 to demonstrate the ability of high-speed grease-lubricated ball bearings to operate satisfactorily through a number of regreasing intervals (300 hours) and with the shaft axis horizontal. The results of this program are presented as an appendix to this report.

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*Boeing Vertol Report D301-10058-1, Rev. A, TEST PLAN--HLH/ATC ENGINE SHAFT SUPPORT BEARING DEVELOPMENT TEST, November 1971.

TECHNICAL APPROACH

BACKGROUND

This development program was conducted to study the capability of a grease-lubricated ball bearing to operate at high speed (632,000 DN) in a simulated helicopter shaft support application. The application selected for test was the HLH engine shaft support bearing operating at a speed of 11,500 rpm. Because of the difficulty of providing fluid lubrication to this bearing, the bearing is designed to be grease lubricated. Another objective of this program was to achieve a regreasing interval of approximately 300 hours.

Given the application, design studies were conducted to determine the space available to support and provide lubrication to this bearing. These studies showed that a basic bearing size of 111-KS could be used and supported in a housing, with the approximate dimensions shown in Figure 1. This housing would be attached to the aircraft structure through two flexible mounts. These mounts would minimize the misalignment across the bearing. In addition, the expected operating conditions for this bearing were defined as follows:

Shaft speed	11,500 rpm
Shaft misalignment	0° 15'
Shaft inclination	34°
Shaft weight	80 pounds
Shaft unbalance	0.10 ± 0.02 inch-ounce
Ambient temperature	180°F

A computer analysis was conducted using the 111-KS ball bearing operating under the above conditions. This initial study (Figure 2) showed that the bearing should have a minimum mounted internal clearance of 0.0009 inch and a maximum of 0.0013 inch in order to reduce friction torque, minimize race depth requirement, and achieve the minimum required B-10 life of 3600 hours (using a material factor of 5 for M-50 steel).

Under subcontract, Marlin-Rockwell Division of TRW, Inc. (referred to as MRC) conducted the development test of the HLH engine shaft support bearing. Testing was initially conducted to determine the optimum housing and bearing modification to achieve a desired regreasing interval of 300 hours. Test results from the first series of tests showed that the major development problems would be improving the grease retention of the housing and providing for a cage design which could operate under these conditions without severe wear or breakage.

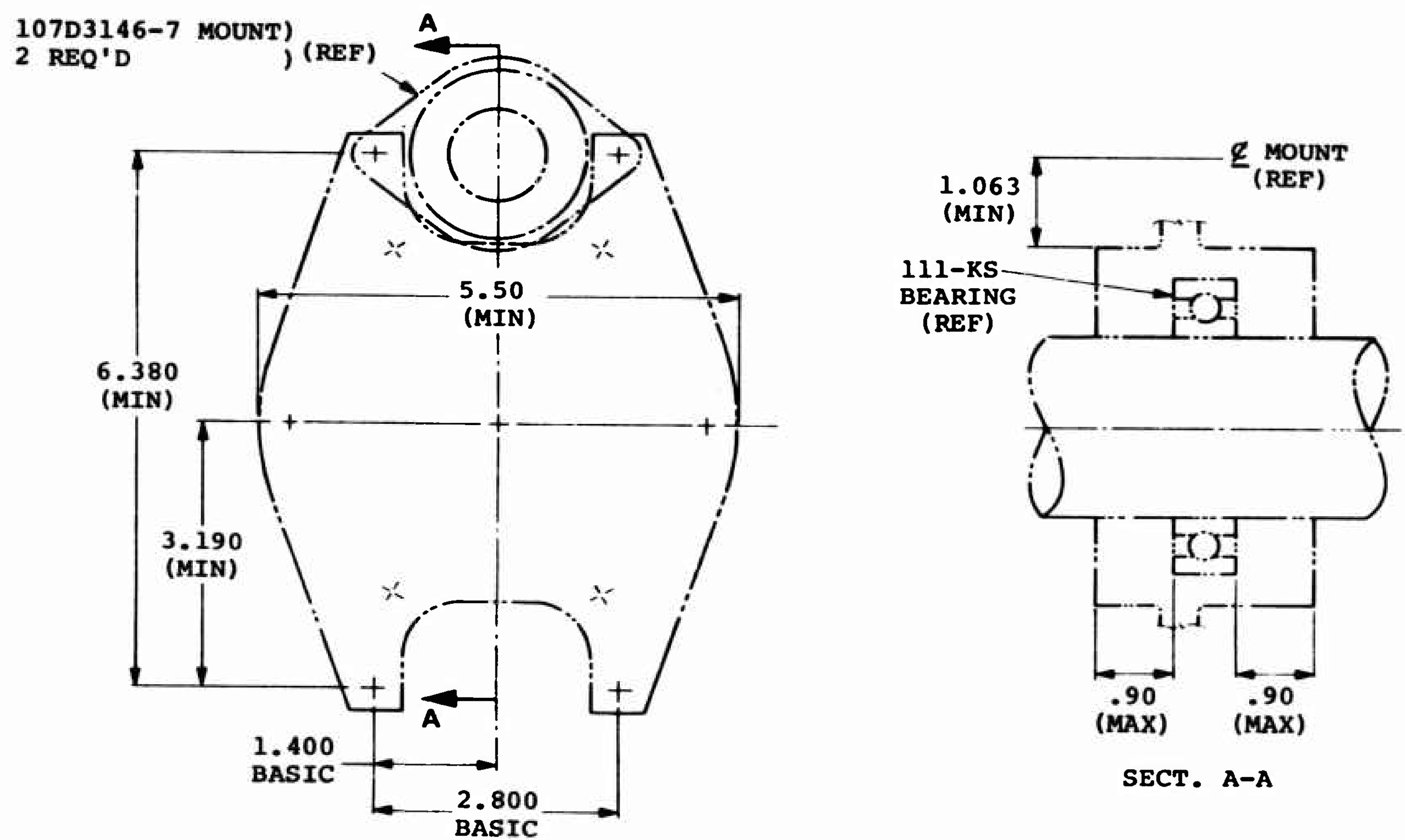


Figure 1. Bearing Housing Envelope Dimensions for HLH Engine Shaft Support Bearing Development Test (SK301-10291).

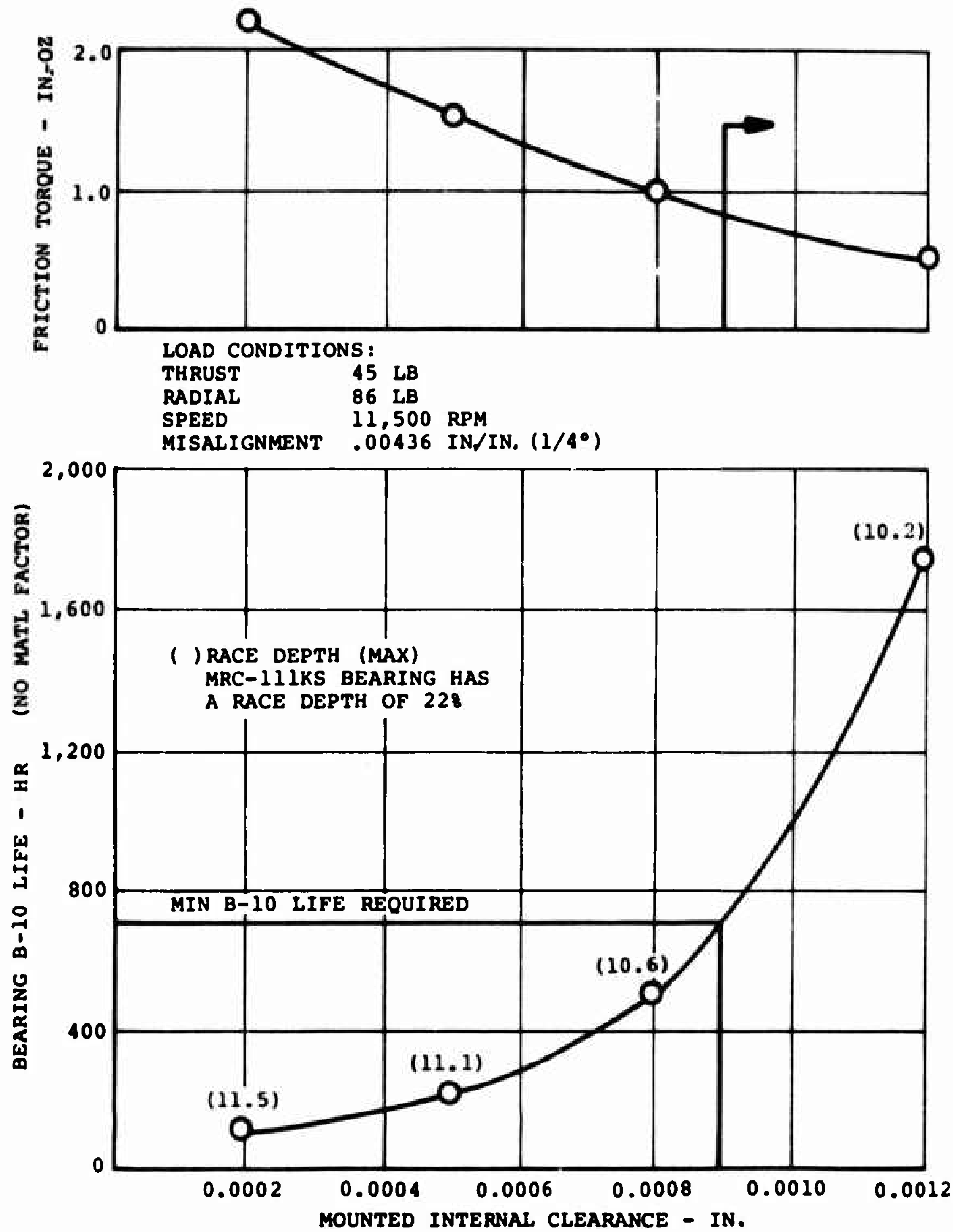


Figure 2. HLH Engine Shaft Support Bearing Study (MRC-111KS Ball Bearing).

As testing proceeded, and as components were examined after test, the initial test data showed that shaft misalignment was greatly affecting the cage performance, from the standpoint of severe land wear and cage breakage. Many attempts were made to reduce this wear by changing ball pocket clearances and cage land clearance. Each of these changes provided some improvement, but none provided satisfactory operation. Therefore, additional computer studies were conducted in an attempt to determine bearing geometry changes which would reduce the effects of shaft misalignment. These studies showed that changes in race curvatures had a significant effect on bearing performance. The results of this study are shown in Table I.

The study showed that increasing the race curvatures to 53% of the ball diameter on the inner race and 54% of the ball diameter on the outer race provided a greater facility for the bearing to accommodate shaft misalignment, and also greatly reduced the internal loading, friction torque, and contact angle variation. Each of these factors should then have a significant effect on improving bearing performance.

Running conditions, especially speed relative to bearing size, usually determine the permissible interval between regreasings. Other factors will also influence the regreasing interval of a bearing. The housing must be designed to allow for maximum grease capacity and at the same time have provisions to insure that the retained grease can be efficiently used. Large quantities of grease trapped in areas away from the critical bearing surfaces will not extend the operating time. Therefore, flingers or recirculating devices may have to be designed into the housing to insure the effective use of all the grease supply on the housing.

A final factor that must be considered in designing a grease-lubricated bearing assembly is to have provisions for facilitating relubrication of the bearing. The housing should be designed so that the new grease supply flow path is on one side of the bearing and the old grease escapes on the other side of the bearing. This insures that the bearing surface is cleaned of old grease and debris and also has an initial supply of new grease. In addition, the housing could be designed so that open space remains in the housing after regreasing. This space will allow the grease to purge during the initial startup; consequently, the risk of possible overlubrication and overheating of the bearing assembly because of too much lubrication should be avoided.

Each of the above design parameters was considered during this development program. Tests were conducted on several housing configurations and bearing designs to achieve the maximum regreasing interval at the HLH operating conditions. Presently, testing is the only method available for evaluating the performance of grease-lubricated bearings.

TABLE I. EFFECT OF VARIATION IN RACE CURVATURES--HLH ENGINE SHAFT SUPPORT BEARING (MRC 111-KS)

Parameter	Design No. 1	Design No. 2	Design No. 3
Speed (rpm)	11,500	11,500	11,500
Thrust Load (lb)	45	45	45
Radial Load (lb)	86	86	86
Misalignment (deg)	1/4	1/4	1/4
Race Curvatures (pct)			
Inner	52	53	54
Outer	52	54	56
Maximum Ball Load (lb)	131	48	42
Mean Hertz Stress (psi)	160,000	124,800	126,540
Ball Path Height (% ball dia)	10.2	6.9	5.5
Max SV Value	299,000	172,700	141,000
Variation in Contact Angle (deg)	46.2	30.5	21.1
Fatigue Life (hr)	1,759	24,100	23,338
Reaction Moment (in.-lb)	261	67	55
Friction Torque (in.-lb)	.55	.10	.08

STATEMENT OF PROBLEMS

The use of grease as a lubricant for ball bearings has usually been limited to speeds of 450,000 DN or less. The HLH/ATC engine shaft support bearing application will require a 111 Conrad type ball bearing to operate at 11,500 rpm (632,000 DN) with grease lubrication.

The test program was divided into several potential problem areas for investigation:

- Lubricant evaluation
- Housing configuration
- Bearing internal geometry
- Bearing cage configuration

The test program was to be conducted so that each of the above problem areas was tested and evaluated to establish the most desirable criteria for achieving a design which would operate successfully for 300 hours without greasing and without any major bearing distress.

The results of these tests will be used to establish the final design of the bearing and housing to be used in the HLH/ATC test program.

TEST METHOD

DESCRIPTION OF TEST SPECIMENS

The MRC 111-KS single-row deep-groove bearing was selected for all full-scale testing. The basic characteristics of this bearing are:

Bore	55 mm
O.D.	90 mm
Width	18 mm
No. of Balls	13
Ball Size	13/32 inch
Ring and Ball Material	M-50 tool steel
Hardnesses	Rockwell C 61.5 to 62.5
ABEC Grade	5
Radial Clearance	0.0016 to 0.0020 inch
Cage Design and Material	Variable

Initially, four cage designs were selected for evaluation:

1. Standard design pressed-steel cage (Figure 3)

2. Pressed-steel cage with Teflon-S coating (Figure 4)

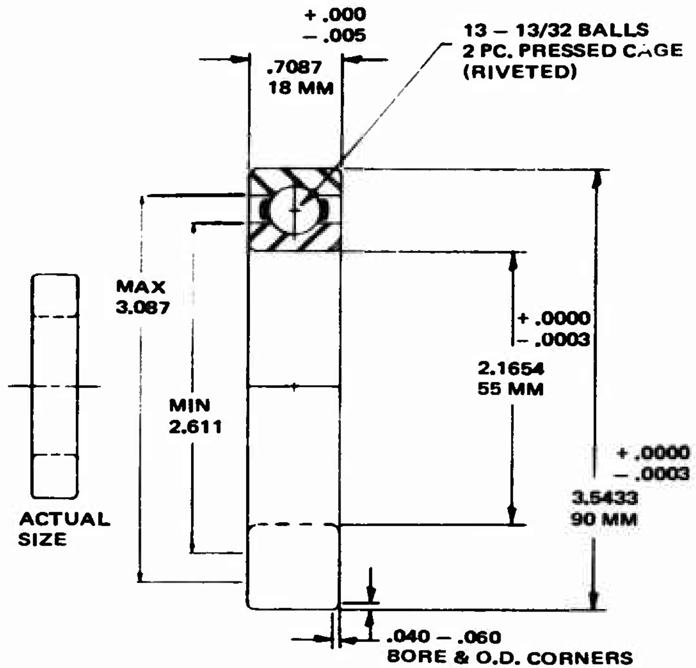
3. Machined inner-land-riding S-Monel cage, Design No. 1 (Figure 5)

4. Fail-safe machined cage with SP-21 inserts (Figure 6)

The standard pressed-steel cage is commonly used in grease-lubricated bearings because its configuration and minimal volume permit a maximum grease fill. It is not usually incorporated in high-speed bearings because of tolerance and manufacturing control. The bearings with these cages were used to establish a baseline data point for all additional tests.

A Teflon-S coating was applied to pressed-steel cages to improve their operation under marginal lubrication. Teflon-S is bonded directly to steel and baked. B and B Plastics, Lockport, New York, applied this approximately 0.001-inch-thick coating.

The original S-Monel inner-land-riding cages (Design 1, Figure 7) were designed to simulate a reduced section cast configuration. They were, however, fully machined because time would not permit fabrication of molds and dies. This design involved thin sections to cut down space requirements and provide room for grease.



(1) ABEC – 5 GRADE

(2) MATERIAL:
RINGS – VACUUM CONSUMABLE
ELECTRODE PROCESS M-50
STEEL R_C 60 MIN.
BALLS – VACUUM CONSUMABLE
ELECTHODE PROCESS M-50
STEEL R_C 60 MIN.
CAGE – SAE 1010 STEEL
RIVETS – SAE 1010 STEEL

(3) RADIAL CLEARANCE .0016–.0020
MEASURED UNDER 11 LBS GAGE LOAD

(4) RIVET SHANK DIA .059

(5) RACE DEPTH:
INNER 22% OF BALL DIA NOM
OUTER 20.5% OF BALL DIA NOM

(6) SURFACE FINISH:

BALLS	–	2 RMS
RACEWAYS	–	8 RMS
O.D.	–	16 RMS
BORE	–	16 RMS
RING FACES	–	16 RMS
RING RIDING LANDS	–	16 RMS
CAGE RIDING LANDS	–	32 RMS

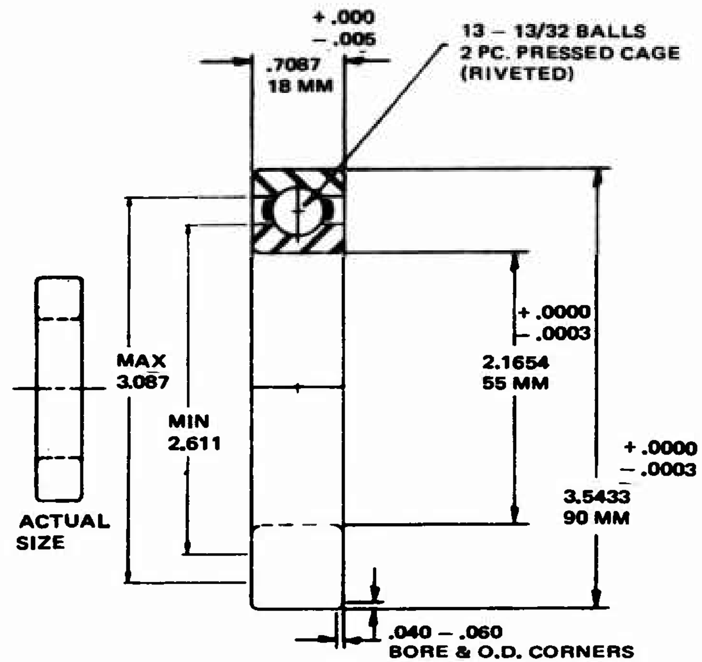
(7) RINGS & BALLS TO BE 100% NITAL ETCH
INSPECTED PER ES-109

(8) RINGS TO BE 100% MAGNETIC PARTICLE
INSPECTED PER ES-105

(9) BEARINGS TO BE 100% VISUAL INSPECTED
PER ES-139

(10) CAGE TO BE 100% FLUORESCENT PENE-
TRANT INSPECTED PER ES-106

Figure 3. Standard Design Pressed-Steel Cage.



(1) ABEC – 5 GRADE

(2) MATERIAL:
- RINGS – VACUUM CONSUMABLE ELECTRODE PROCESS M-50 STEEL R_C 60 MIN.
- BALLS – VACUUM CONSUMABLE ELECTRODE PROCESS M-50 STEEL R_C 60 MIN.
- CAGE – SAE 1010 STEEL WITH TEFLON COATING ON BALL POCKETS .001–.002 THICK
- RIVETS – SAE 1010 STEEL

(3) RADIAL CLEARANCE .0016–.0020 MEASURED UNDER 11 LBS GAGE LOAD

(4) RIVET SHANK DIA .059

(5) SURFACE FINISH:

BALLS	–	2 RMS
RACEWAYS	–	8 RMS
O.D.	–	16 RMS
BORE	–	16 RMS
RING FACES	–	16 RMS
RING RIDING LANDS	–	16 RMS
CAGE RIDING LANDS	–	32 RMS

(6) RACE DEPTH:
- INNER 22% OF BALL DIA NOM
- OUTER 20.5% OF BALL DIA NOM

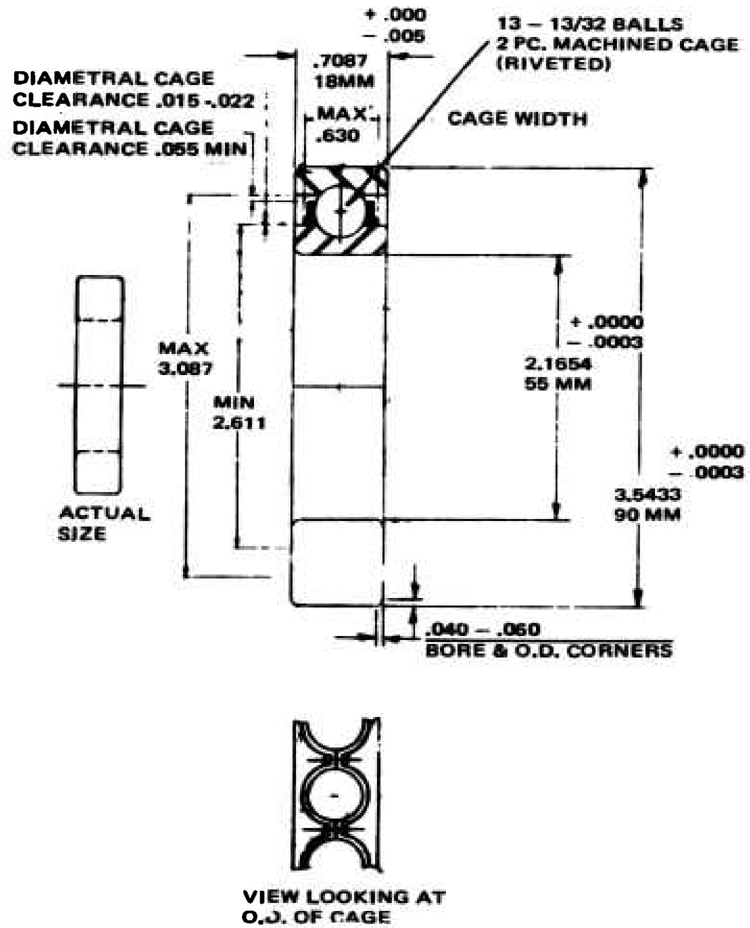
(7) RINGS TO BE 100% MAGNETIC PARTICLE INSPECTED PER ES-105

(8) BEARINGS TO BE 100% VISUAL INSPECTED PER ES-139

(9) CAGE MUST BE FLUORESCENT PENETRANT INSPECTED PER ES-106

(10) RINGS TO BE 100% NITAL ETCH INSPECTED PER ES-109

Figure 4. Pressed Steel Cage With Teflon-S Coating.



(1) ABEC – 5 GRADE

(2) MATERIAL:
RINGS – VACUUM CONSUMABLE ELECTRODE PROCESS M-50 STEEL R_C 60 MIN.
BALLS – VACUUM CONSUMABLE ELECTRODE PROCESS M-50 STEEL R_C 60 MIN.
CAGE – S-MONEL
RIVETS – SAE 1010 STEEL

(3) RADIAL CLEARANCE .0016–.0020 MEASURED UNDER 11 LBS GAGE LOAD

(4) SURFACE FINISH:

BALLS	– 2 RMS
RACEWAYS	– 8 RMS
O.D.	– 16 RMS
BORE	– 16 RMS
RING FACES	– 16 RMS
RING RIDING LANDS	– 16 RMS
CAGE RIDING LANDS	– 32 RMS

(5) RACE DEPTH:
INNER 22% OF BALL DIA NOM
OUTER 20.5% OF BALL DIA NOM

(6) RINGS TO BE 100% MAGNETIC PARTICLE INSPECTED PER ES-05

(7) BEARINGS TO BE 100% VISUAL INSPECTED PER ES-139

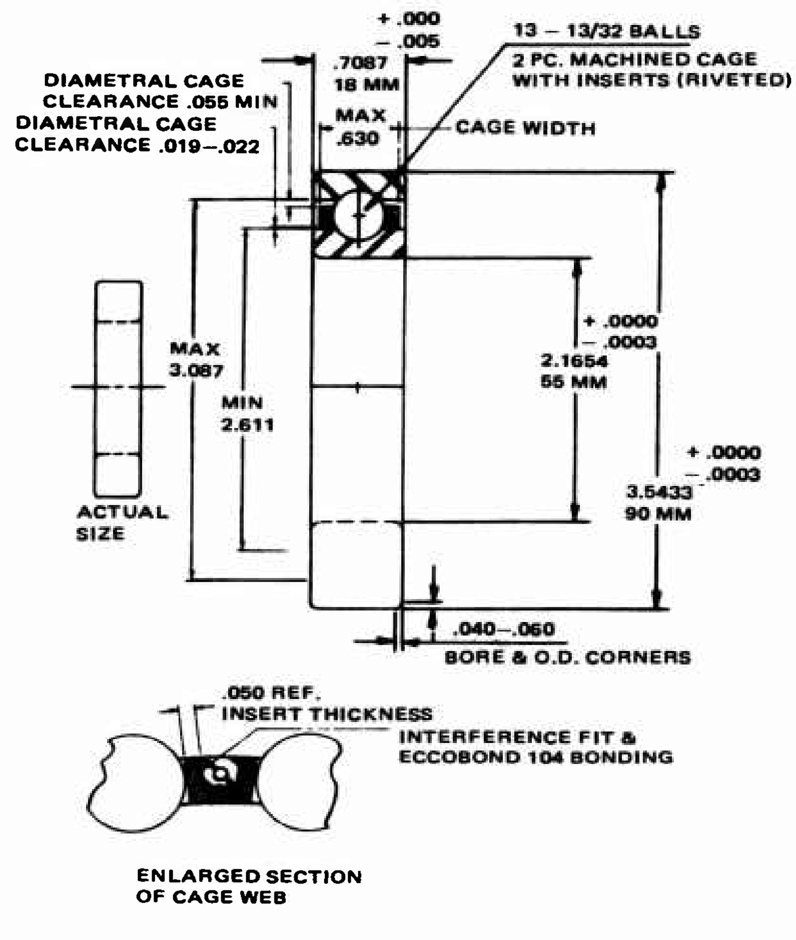
(8) CAGE MUST BE 50% FLUORESCENT PENETRANT INSPECTED PER ES-106

(9) RINGS & BALLS TO BE 100% NITAL ETCH INSPECTED PER ES-109

(10) RIVET SHANK DIA .051

(11) CAGE POCKET CLEARANCE .023–.028

Figure 5. Machined Inner-Land-Riding S-Monel Cage, Design 1.



(1) ABEC – 5 GRADE

(2) MATERIAL:
RINGS – VACUUM CONSUMABLE
ELECTRODE PROCESS M-50
STEEL R_C 60 MIN
BALLS – VACUUM CONSUMABLE
ELECTRODE PROCESS M-50
STEEL R_C 60 MIN
CAGE – SAE 4340 STEEL
INSERTS – SP-21
RIVETS – SAE 1010 STEEL

(3) RADIAL CLEARANCE .0016–.0020
MEASURED UNDER 11 LBS GAGE LOAD

(4) RIVET SHANK DIA .051

(5) CAGE POCKET CLEARANCE .023–.028

(6) SURFACE FINISH:

BALLS	– 2 RMS
RACEWAYS	– 8 RMS
O.D.	– 16 RMS
BORE	– 16 RMS
RING FACES	– 16 RMS
RING RIDING LANDS	– 16 RMS
CAGE RIDING LANDS	– 32 RMS

(7) RACE DEPTH:
INNER 22% OF BALL DIA NOM
OUTER 20.5% OF BALL DIA NOM

(8) RINGS TO BE 100% MAGNETIC PARTICLE
INSPECTED PER ES-106

(9) BEARINGS TO BE 100% VISUAL INSPECTED
PER ES-139

(10) CAGE MUST BE 100% FLUORESCENT PENETRANT
INSPECTED PER ES-106

(11) RINGS TO BE 1007, NITAL ETCH INSPECTED
PER ES-107

Figure 6. Fail-Safe Machined Cage With SP-21 Inserts.

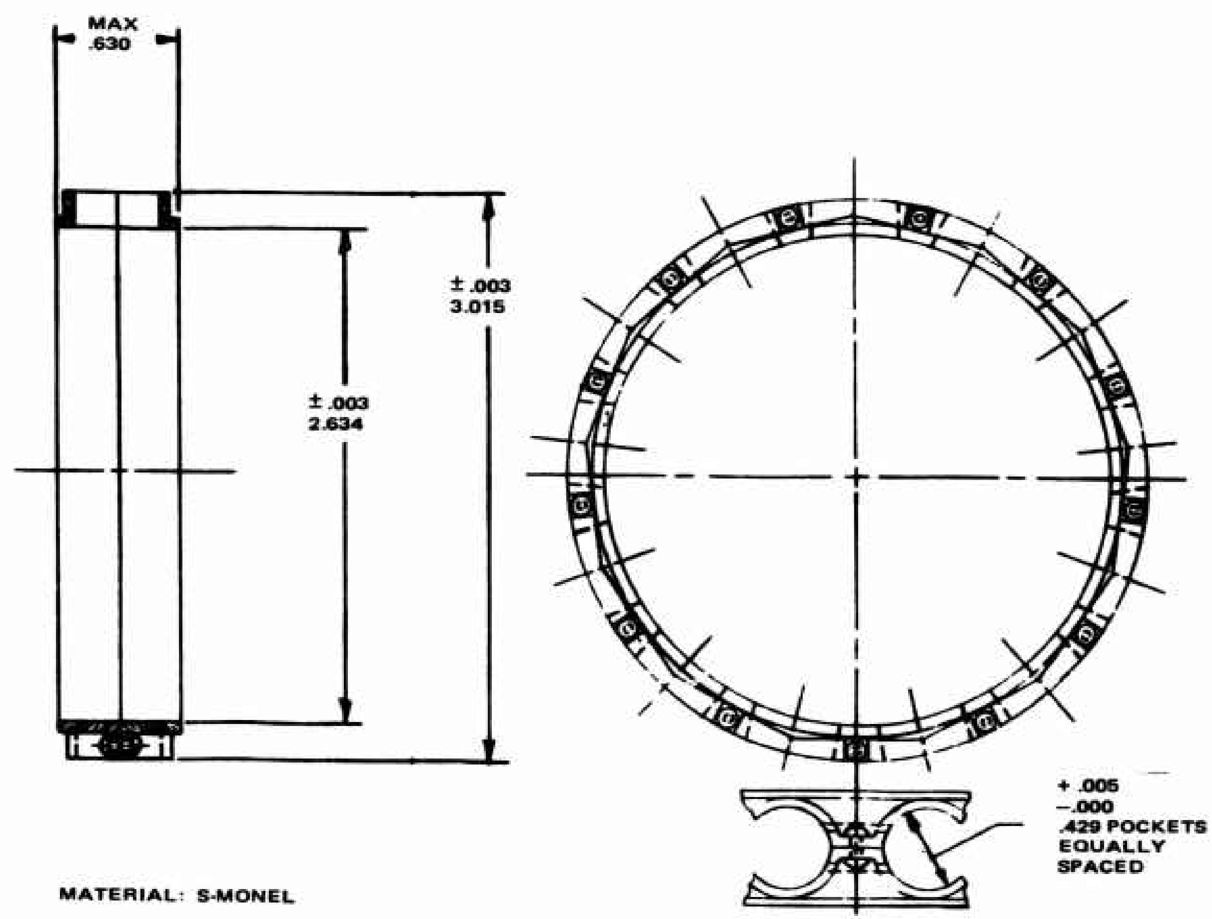


Figure 7. Inner-Land-Riding S-Monel Cage, Design 1.

When this design showed promise, but demonstrated some limitations, additional inner-land-riding S-Monel designs were made to provide more clearance for grease movement. Final designs involved greatly reduced sections and much more clearance between lands and land riding surfaces to evaluate this approach.

As testing progressed, the following additional cage designs were added to the program (although not necessarily in the following order):

- Machined inner-land-riding S-Monel cage, Design No. 2 (Figure 8)
- Machined inner-land-riding S-Monel cage, Design No. 3 (Figure 9)
- Machined inner-land-riding S-Monel cage, Design No. 4 (Figure 10)
- Machined inner-land-riding S-Monel cage, Design No. 5 (Figure 11)
- Machined outer-land-riding S-Monel cage (Figure 12)
- Modified insert type cage (rework design)
- Two-piece machined steel cage, silver-plated, inner-land-riding (Figure 13)

Because of continuing difficulty in achieving a successfully operating bearing for 300 hours, two MRC 111-KS ball bearings conforming to Figure 5 were obtained and modified as shown on Figure 14. The radius of curvature of the inner races was changed to 53% of ball diameter, and that of the outer races was changed to 54% of ball diameter. The raceway curvatures used in previous testing were 52% of ball diameter. One inner ring was coated with Teflon-S on its land surfaces. Cages conformed to Design 3 of the inner-land-riding S-Monel configuration.

The fail-safe cage designed with DuPont's SP-21 inserts followed design principles which had been shown to permit bearing operation for a period of time after lubricating oil shutoff. In effect, the low friction inserts (machined from graphite-filled polymide slugs) provided dry lubrication after oil depletion. In the current application, it was felt that the concept might supplement the grease.

Figures 15 through 20 show the sequence of fabricating the insert type cages.

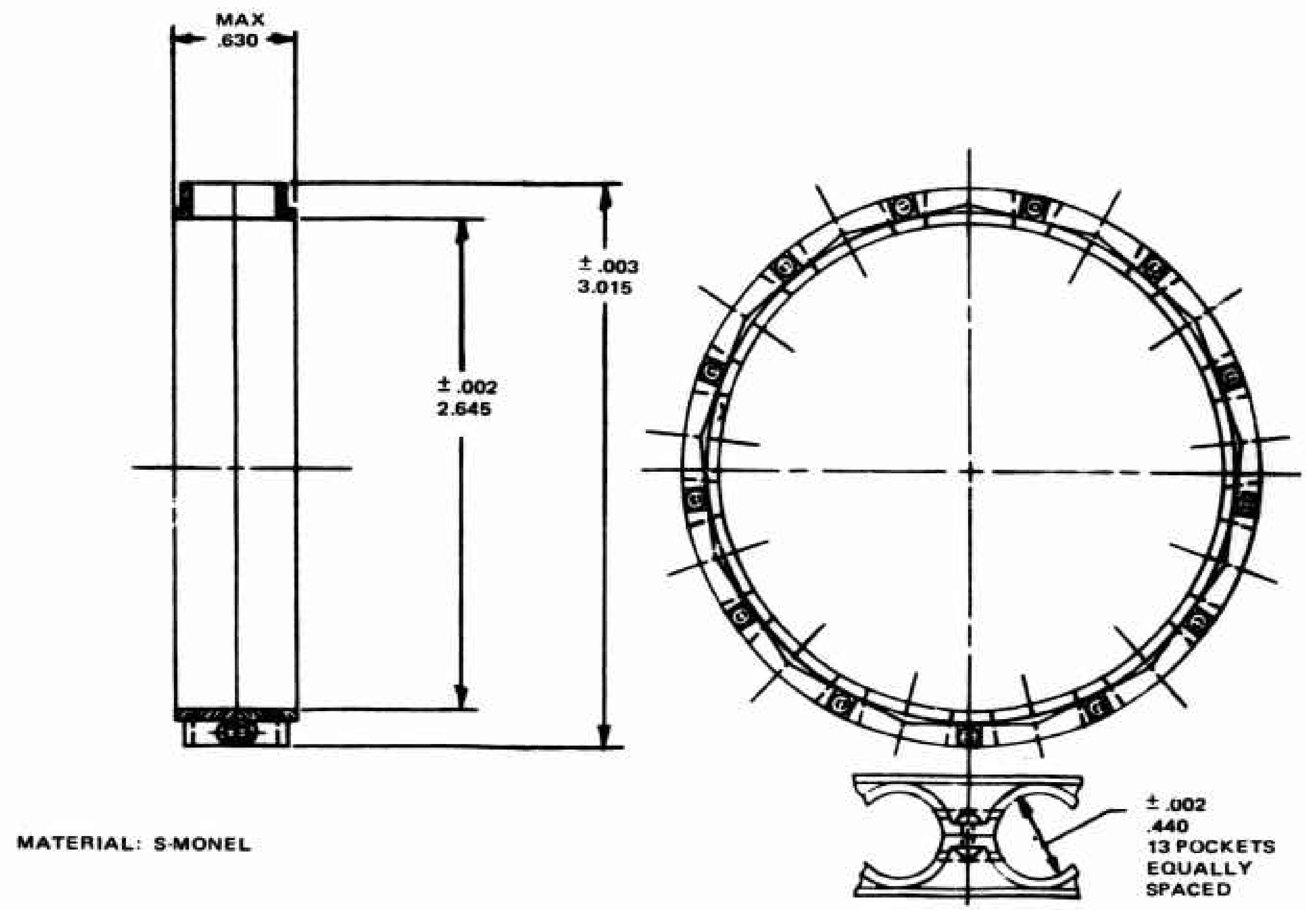


Figure 8. Machined Inner-Land-Riding S-Monel Cage, Design 2.

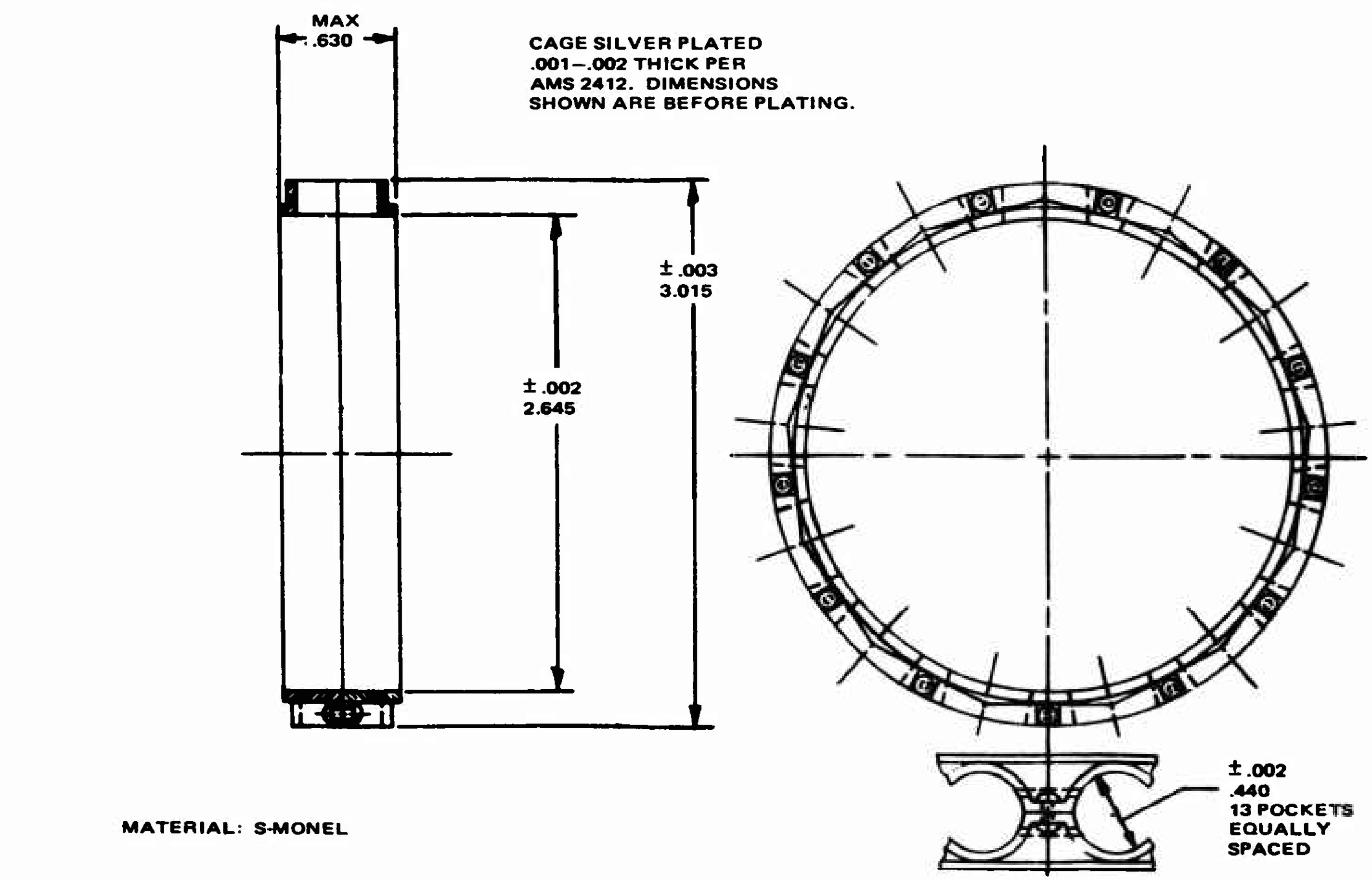


Figure 9. Machined Inner-Land-Riding S-Monel Cage, Design 3.

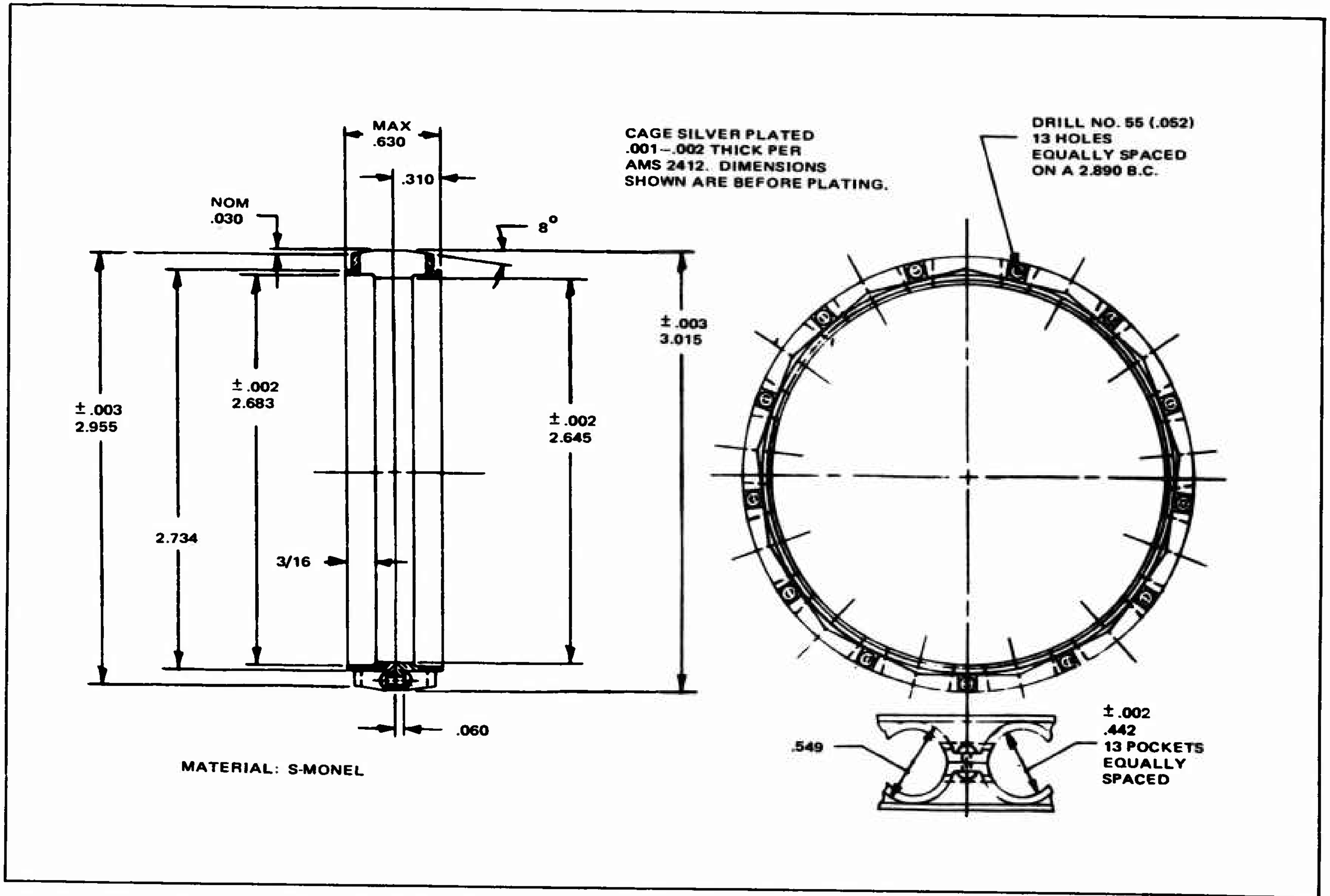


Figure 10. Machined Inner-Land-Riding S-Monel Cage, Design 4.

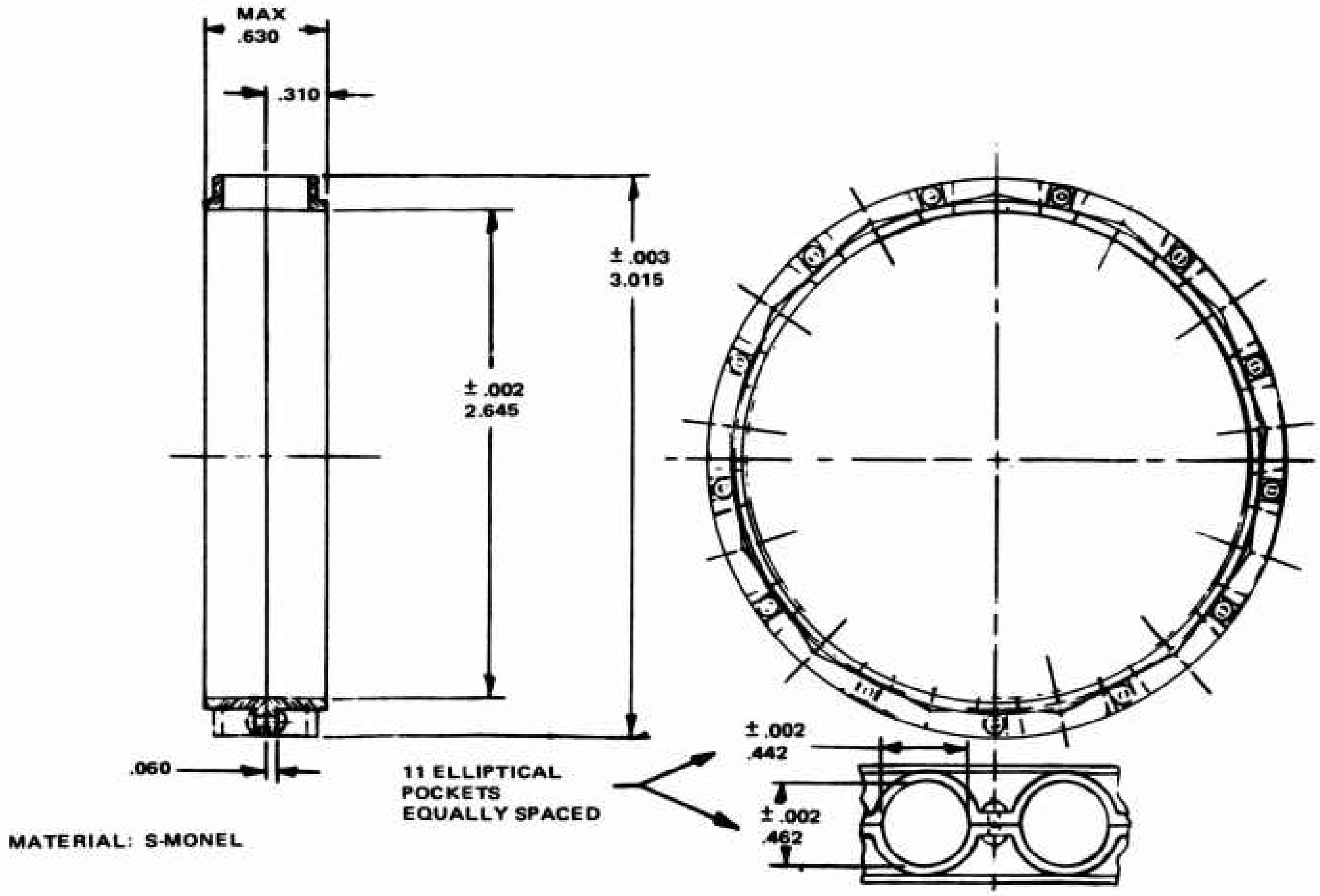


Figure 11. Machined Inner-Land-Riding S-Monel Cage, Design 5.

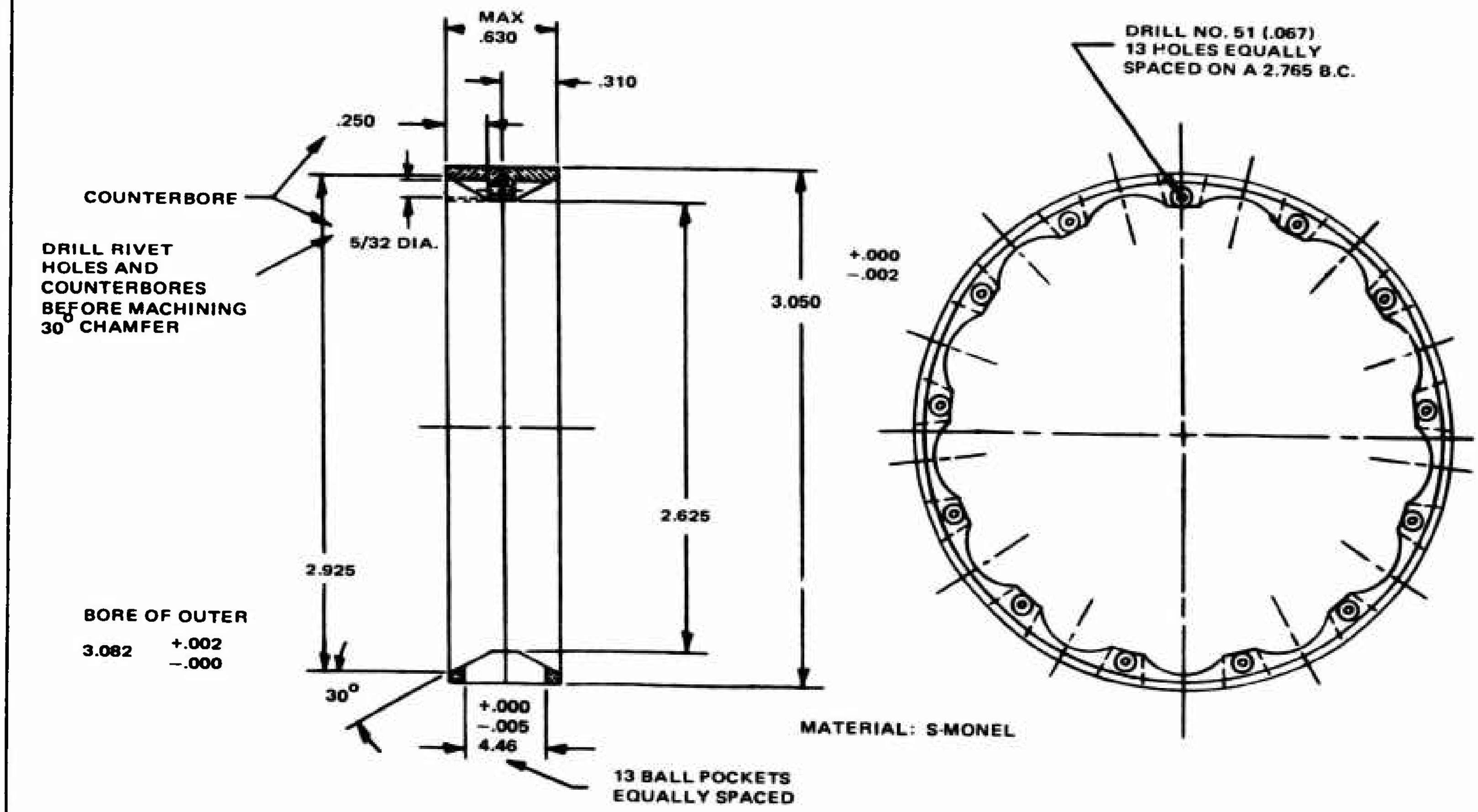


Figure 12. Machined Outer-Land-Riding S-Monel Cage.

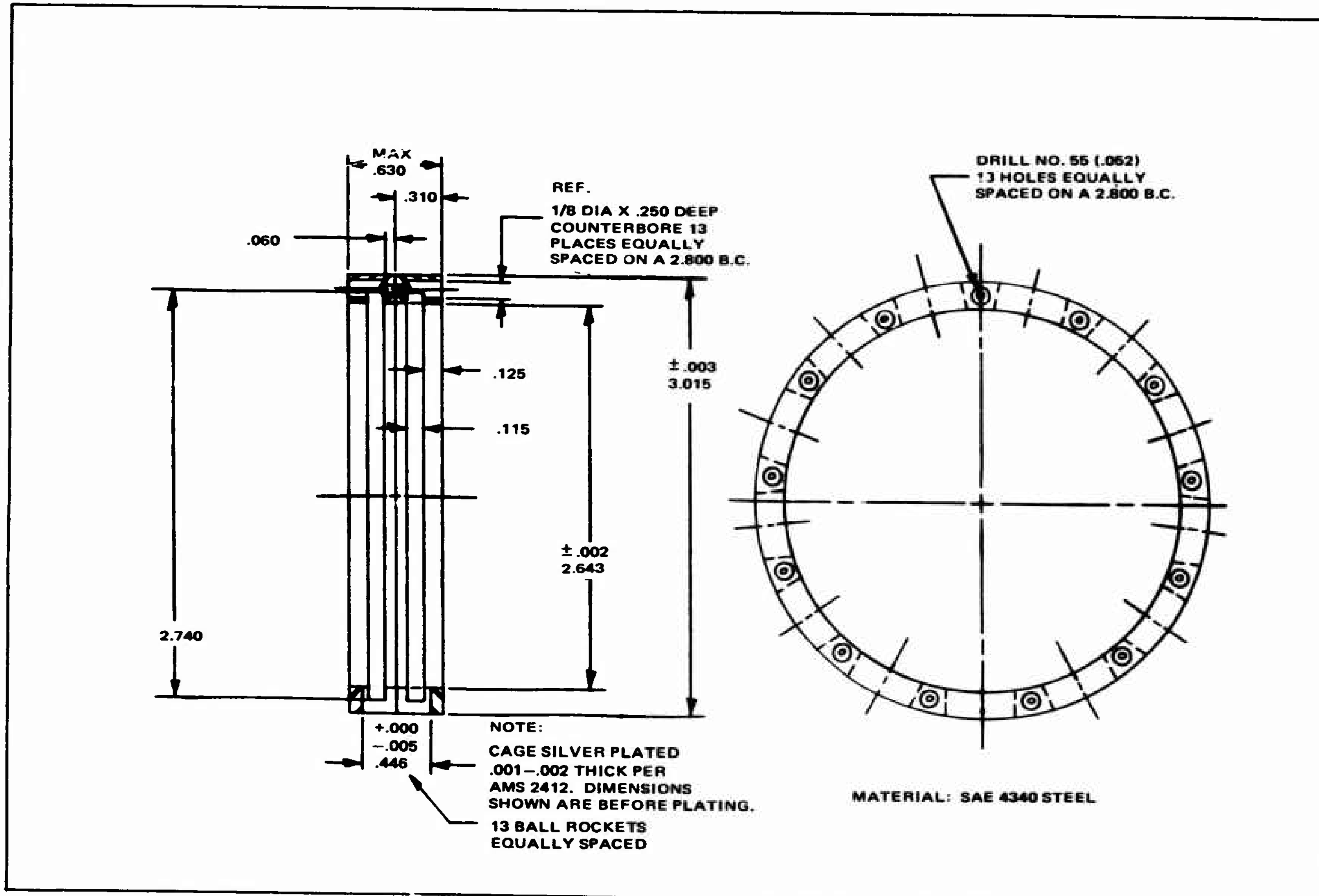
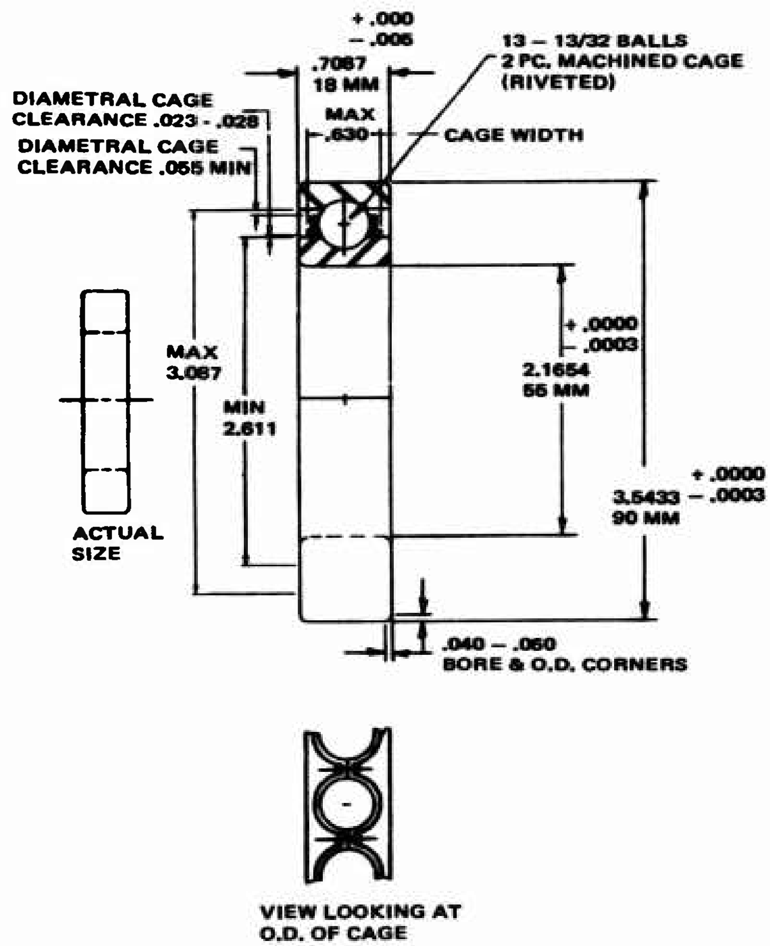


Figure 13. Two-Piece Machined Steel Cage, Silver Plated.



(1) ABEC – 5 GRADE

(2) MATERIAL:
RINGS – VACUUM CONSUMABLE ELECTRODE PROCESS M-50 STEEL R_C 60 MIN.
BALLS – VACUUM CONSUMABLE ELECTRODE PROCESS M-50 STEEL R_C 60 MIN.
CAGE – S-MONEL
RIVETS – SAE 1010 STEEL

(3) RADIAL CLEARANCE .0016–.0020 MEASURED UNDER 11 LBS GAGE LOAD

(4) SURFACE FINISH:

BALLS	– 2 RMS
RACEWAYS	– 8 RMS
O.D.	– 16 RMS
BORE	– 16 RMS
RING FACES	– 16 RMS
RING RIDING LANDS	– 16 RMS
CAGE RIDING LANDS	– 32 RMS

(5) RACE DEPTH:
INNER 22% OF BALL DIA NOM
OUTER 20.5% OF BALL DIA NOM

(6) RINGS TO BE 100% MAGNETIC PARTICLE INSPECTED PER ES-106

(7) BEARINGS TO BE 100% VISUAL INSPECTED PER ES-139

(8) CAGE MUST BE 100% FLUORESCENT PENETRANT INSPECTED PER ES-106

(9) RINGS & BALLS TO BE 100% NITAL ETCH INSPECTED PER ES-109

(10) RIVET SHANK DIA .051

(11) CAGE POCKET CLEARANCE .031–.036

(12) CAGE SILVER PLATED .001–.002 THICK PER AMS 2412

Figure 14. Modified 111-KS Test Bearing.

Figure 15. Vespel SP-21 Inserts.

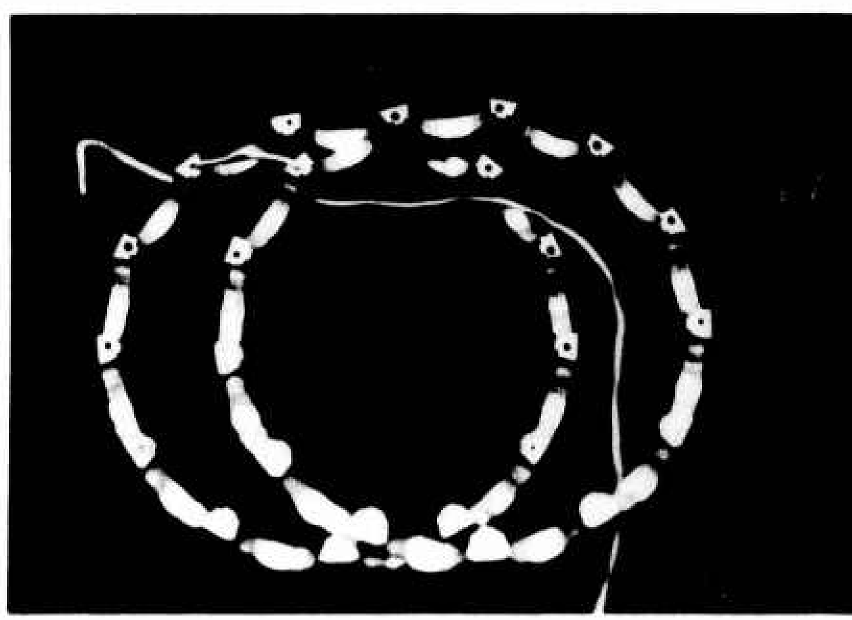


Figure 16. Steel Frame for Insert Type Cage.

Figure 17. Inserts Fitted Into Frame.

Figure 18. Inserts Glued Into Frame.

Figure 19. Cage in Finish-Ground Condition.

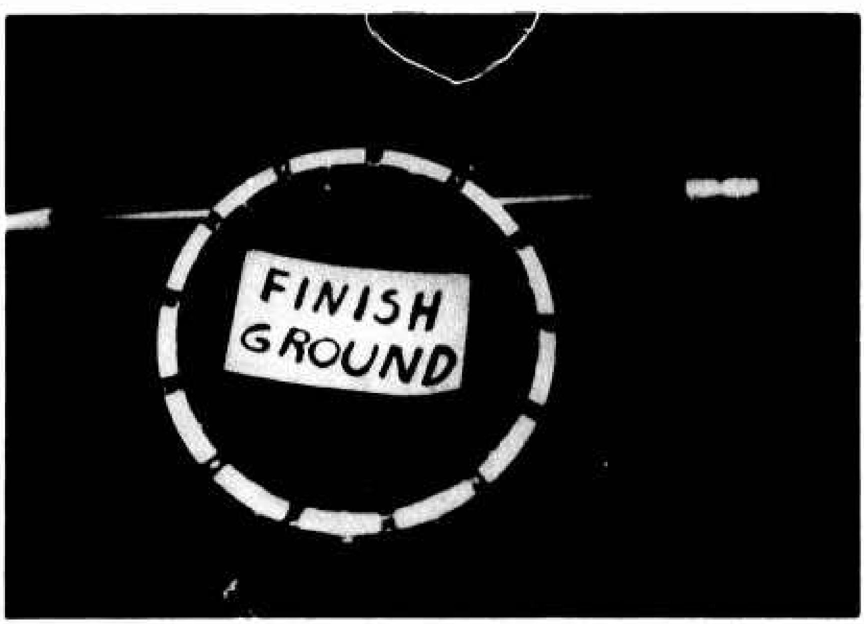


Figure 20. Cage in Finish-Ground Condition.

Testing of the first insert type cages indicated insufficient lubrication. Two of these cages were then reworked as shown in Figure 21 to reduce stock and thereby increase grease fill and internal grease circulation.

An outer land riding S-Monel cage (Figure 12) was designed and fabricated after significant wear was observed in the inner land riding specimens. The rationale involved the action of centrifugal force on an unbalanced cage; an inner land riding cage tends to wear only in one area, with rate of wear increasing as the cage becomes more and more unbalanced. An outer land riding cage tends to wear more evenly about its periphery, although other factors, such as the tendency of an outer land riding cage to trap lubricant in the outer race, mitigate against its universal use.

After severe wear, indications of inadequate grease circulation, and fracture of a number of cages occurred during testing, a silver-plated machined steel cage (Figure 13) was designed and fabricated. The rail section and short rivet design were incorporated for high strength; the silver plate provided some marginal lubrication; internal annular grooves provided space for grease circulation around the inner race.

To assist in the lubrication of the inner ring lands, a coating of Teflon-S (Figure 22) was applied to the cage riding lands of a few bearings. The material, applied by B and B Plastics, was approximately 0.001 inch thick. An additional coat of FEP Teflon was applied to half of the treated inner rings.

The cage designs and housing configurations were evaluated and used to establish an apparently successful grease sealing method, along with some bearing design parameters. Additional testing was set up to evaluate a final bearing and housing configuration and to establish some level of reliability assurance by repetitive testing. The series of designs tested during this final phase were as follows:

- Two bearings were fabricated to the configuration shown in Figure 23. The inner race cage riding lands were coated with "Sermetel 72", by Sermetel Division, North Wales, Pennsylvania. The coating was a final attempt to reduce the severe cage land wear experienced during prior testing.

- A major review of all the cage designs tested indicated that the original ball riding, pressed steel cage appeared to operate satisfactorily under the specified test conditions. Therefore, a machined ball riding cage (Figure 24) was designed and fabricated. This final cage design was assembled into the bearing configuration shown in Figure 25.

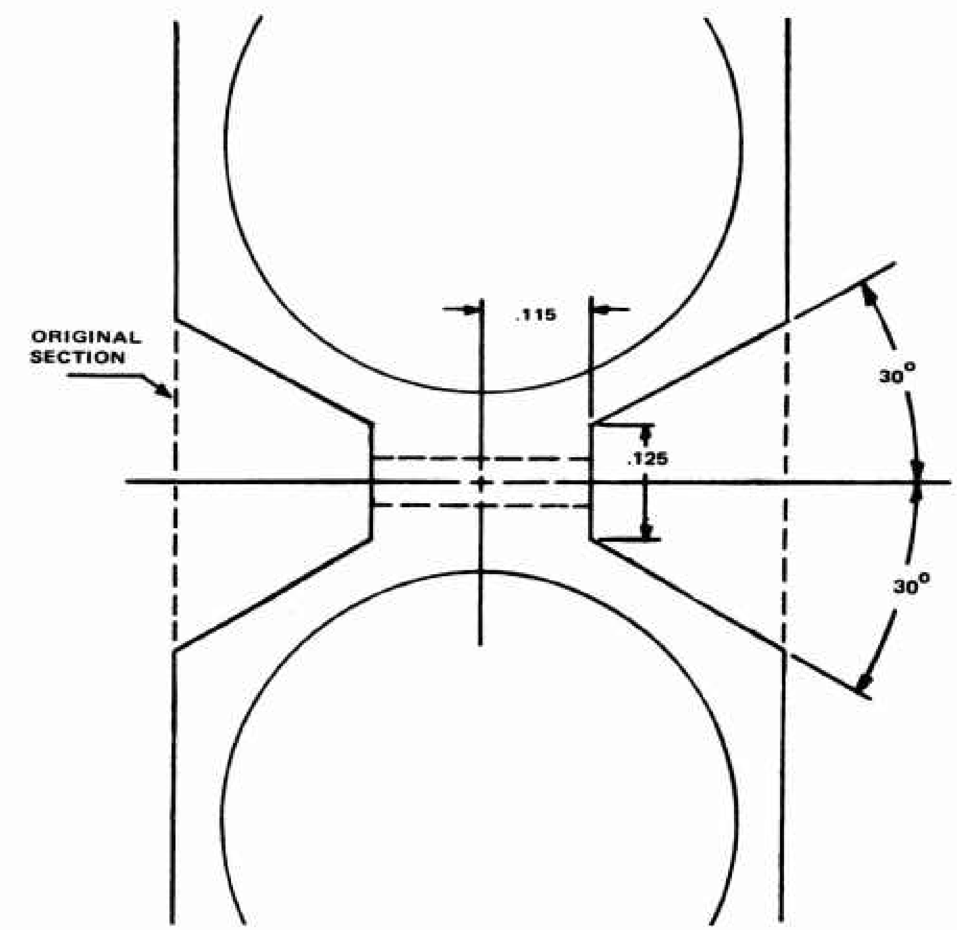
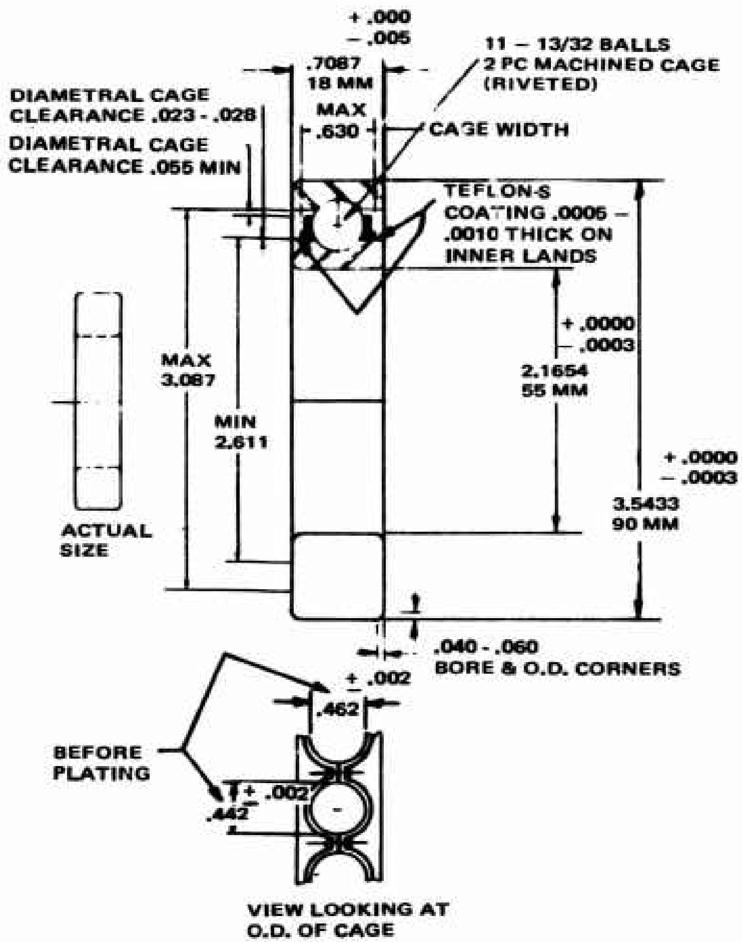


Figure 21. 111-KS Insert Cage Modified.



(1) ABEC − 5 GRADE

(2) MATERIAL:
RINGS − VACUUM CONSUMABLE ELECTRODE PROCESS M-50 STEEL R_C 60 MIN.
BALLS − VACUUM CONSUMABLE ELECTRODE PROCESS M-50 STEEL R_C 60 MIN.
CAGE − S-MONEL
RIVETS − SAE 1010 STEEL

(3) RADIAL CLEARANCE .0016−.0020 MEASURED UNDER 11 LBS GAGE LOAD

(4) SURFACE FINISH:

BALLS	− 2 RMS
RACEWAYS	− 8 RMS
O.D.	− 16 RMS
BORE	− 16 RMS
RING FACES	− 16 RMS
RING RIDING LANDS	− 16 RMS
CAGE RIDING LANDS	− 32 RMS

(5) RACE DEPTH:
INNER 22% OF BALL DIA NOM
OUTER 20.5% OF BALL DIA NOM

(6) RINGS TO BE 100% MAGNETIC PARTICLE INSPECTED PER ES-105

(7) BEARINGS TO BE 100% VISUAL INSPECTED PER ES-139

(8) CAGE MUST BE 100% FLUORESCENT PENETRANT INSPECTED PER ES-106

(9) RINGS & BALLS TO BE 100% NITAL ETCH INSPECTED PER ES-109

(10) RIVET SHANK DIA .066

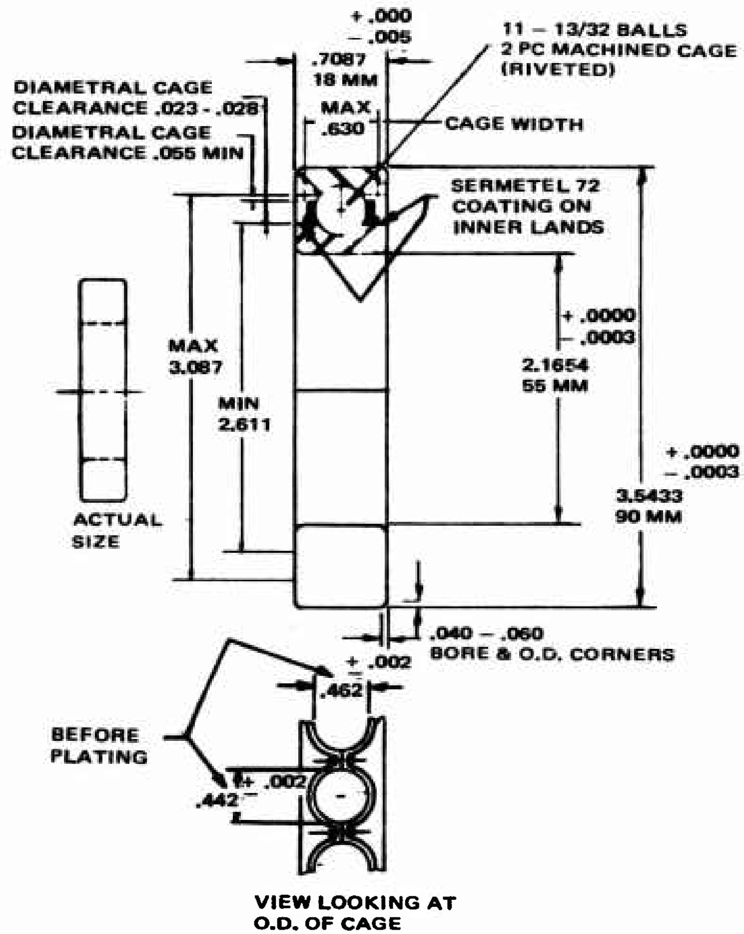
(11) CAGE POCKET CLEARANCE .031−.036

(12) CAGE SILVER PLATED .001−.002 THICK PER AMS 2412

(13) CAGE TO BE BALANCED TO 3 GRAM CENTIMETER MAX

(14) INNER RACE RADIUS 53% OF BALL DIA NOM
OUTER RACE RADIUS 54% OF BALL DIA NOM

Figure 22. Modified 111-KS Test Bearing.



(1) ABEC – 5 GRADE

(2) MATERIAL:
RINGS – VACUUM CONSUMABLE ELECTRODE PROCESS M-50 STEEL R_C 60 MIN.
BALLS – VACUUM CONSUMABLE ELECTRODE PROCESS M-50 STEEL R_C 60 MIN.
CAGE – S-MONEL
RIVETS – SAE 1010 STEEL

(3) RADIAL CLEARANCE .0016–.0020 MEASURED UNDER 11 LBS GAGE LOAD

(4) SURFACE FINISH:

BALLS	–	2 RMS
RACEWAYS	–	8 RMS
O.D.	–	16 RMS
BORE	–	16 RMS
RING FACES	–	16 RMS
RING RIDING LANDS	–	16 RMS
CAGE RIDING LANDS	–	32 RMS

(5) RACE DEPTH:
INNER 22% OF BALL DIA NOM
OUTER 20.5% OF BALL DIA NOM

(6) RINGS TO BE 100% MAGNETIC PARTICLE INSPECTED PER ES-105

(7) BEARINGS TO BE 100% VISUAL INSPECTED PER ES-139

(8) CAGE MUST BE 100% FLUORESCENT PENETRANT INSPECTED PER ES-106

(9) RINGS & BALLS TO BE 100% NITAL ETCH INSPECTED PER ES-109

(10) RIVET SHANK DIA .066

(11) CAGE POCKET CLEARANCE .031–.036

(12) CAGE SILVER PLATED .001–.002 THICK PER AMS 2412

(13) CAGE TO BE BALANCED TO 3 GRAM CENTI-METER MAX

(14) INNER RACE RADIUS 53% OF BALL DIA NOM
OUTER RACE RADIUS 54% OF BALL DIA NOM

Figure 23. Modified Test Bearing, Sermetal 72.

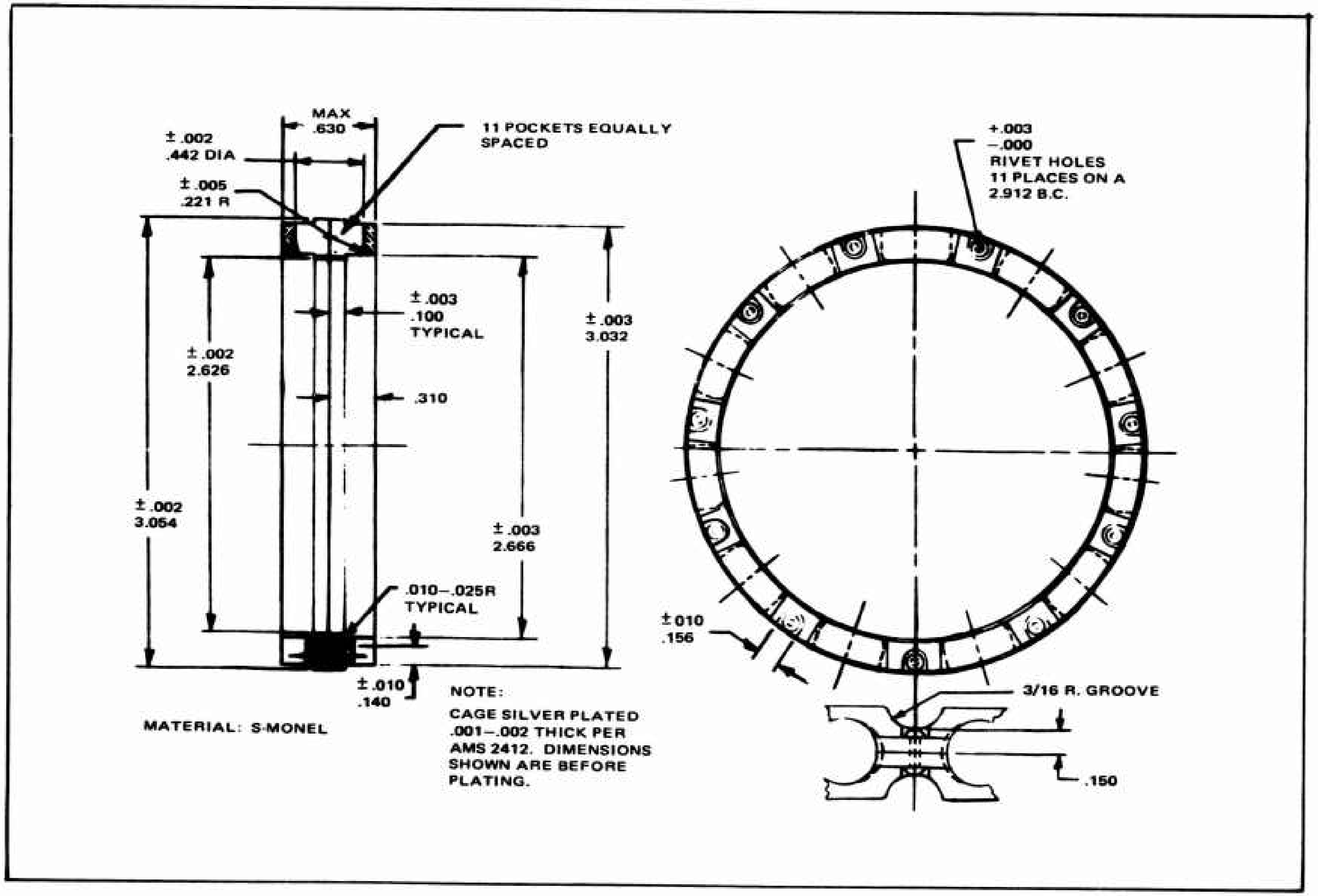
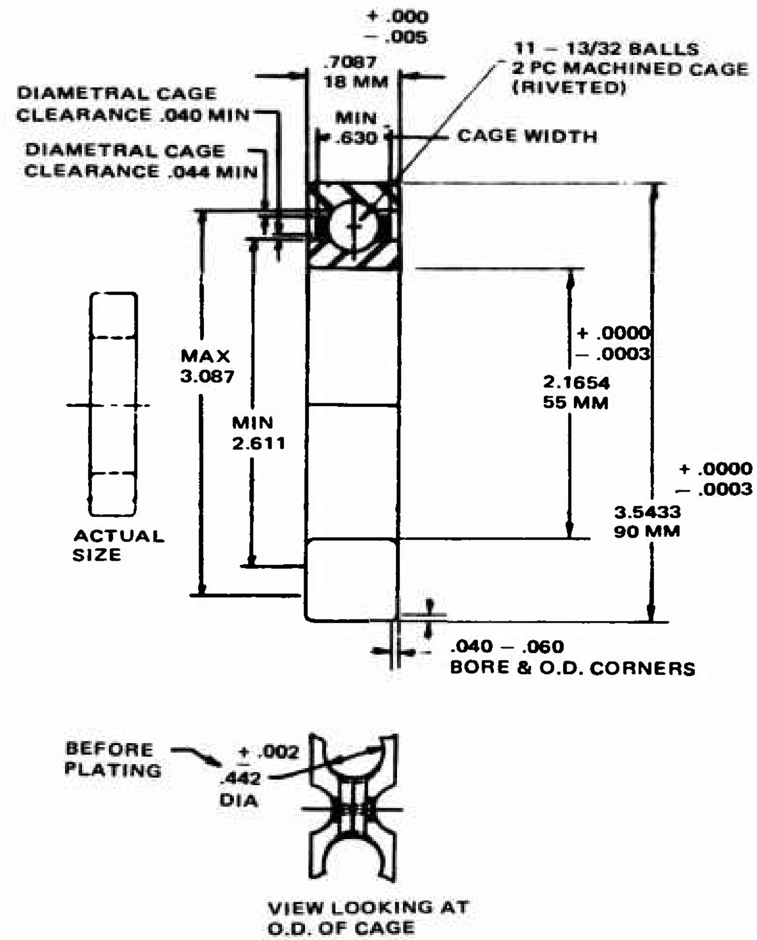


Figure 24. Machine Ball Riding Cage, Design 6.



(1) ABEC – 5 GRADE

(2) MATERIAL:
RINGS – VACUUM CONSUMABLE ELECTRODE PROCESS M-50 STEEL R_C 60 MIN.
BALLS – VACUUM CONSUMABLE ELECTRODE PROCESS M-50 STEEL R_C 60 MIN.
CAGE – S-MONEL
RIVETS – SAE 1010 STEEL

(3) RADIAL CLEARANCE .0016–.0020 MEASURED UNDER 11 LBS GAGE LOAD

(4) SURFACE FINISH:

BALLS	– 2 RMS
RACEWAYS	– 8 RMS
O.D.	– 16 RMS
BORE	– 16 RMS
RING FACES	– 16 RMS
O.D. OF INNER	– 16 RMS
BORE OF OUTER	– 16 RMS

(5) RACE DEPTH:
INNER 22% OF BALL DIA NOM
OUTER 20.5% OF BALL DIA NOM

(6) RINGS TO BE 100% MAGNETIC PARTICLE INSPECTED PER ES-105

(7) BEARINGS TO BE 100% VISUAL INSPECTED PER ES-139

(8) CAGE MUST BE 100% FLUORESCENT PENETRANT INSPECTED PER ES-106

(9) RINGS & BALLS TO BE 100% NITAL ETCH INSPECTED PER ES-109

(10) RIVET SHANK DIA .066

(11) CAGE POCKET CLEARANCE .031–.036

(12) CAGE SILVER PLATED .001–.002 THICK PER AMS 2412

(13) CAGE TO BE BALANCED TO 3 GRAM CENTI-METER MAX.

(14) INNER RACE RADIUS 53% OF BALL DIA NOM
OUTER RACE RADIUS 54% OF BALL DIA NOM

Figure 25. Modified Test Bearing, Cage Design 6.

PRELIMINARY GREASE EVALUATION TEST RIG

Preliminary grease screening testing was conducted using an MRC 1000°F Grease Test Spindle whose basic use is to evaluate the high-temperature performance requirements of most military grease specifications. The test method employs two 204-K Conrad-type bearings on a belt-driven spindle (Figure 26). The unit can be used for testing lubricants at temperatures up to 1000°F, with applied bearing thrust loads from 0 to more than 1000 pounds and at speeds up to 35,000 rpm. All spindle parts are fabricated from stabilized high-temperature die steel, and all critical surfaces are hard chrome-plated to prevent high-temperature oxidation. This test unit meets the requirements for the apparatus specified in the Coordinating Research Council L-54 Research Technique.

The test rig was set up with a high-speed drive (Figure 27) to evaluate the performance and capabilities of candidate greases to lubricate the 204 test bearing at a DN of 640,000. The operating conditions of the test rig were:

Speed	32,000 rpm
Bearing	MRC 204-S-17
Thrust load	50 pounds
Lubrication	Candidate greases
Temperature	180°F or higher, no heat added

Only the outer race temperature of the rear bearing was measured; therefore, failure or deterioration of the front bearing was not indicated by temperature data. Wattmeter data showing motor load variation (changes in test bearing torque) were also used for the purpose of detecting bearing failure.

FULL-SCALE BEARING TEST RIG

Two test machines were fabricated to conduct full-scale bearing tests. The test machines were designed to operate under the following conditions:

Shaft speed	11,500 rpm
Shaft incline angle	34°
Shaft misalignment	15 minutes
Ambient temperature	180°F
Shaft weight	80 pounds
Shaft unbalance	0.10 ± 0.02 inch-ounce

In addition, the bearing housings were required to match the external configuration (as specified in Figure 1), and the housings were to be mounted on elastomeric mounts such as those furnished by Boeing Vertol to MRC. These mounts are similar to those presently used on helicopters to reduce shaft misalignments and improper positioning of the shaft

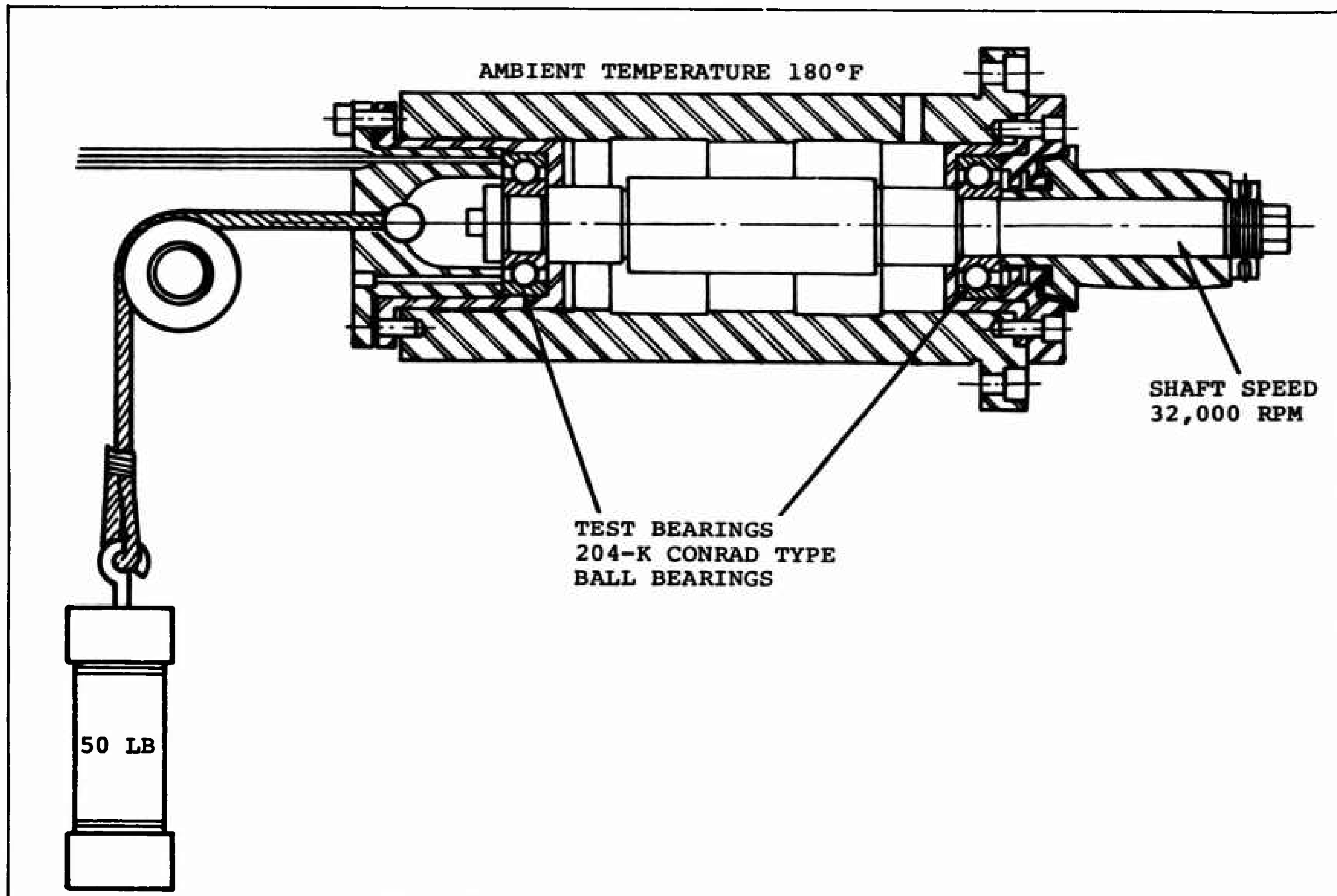


Figure 26. MRC High-Temperature Grease Test Spindle.

Figure 27. Grease Evaluation Test Rig.

and bearing. Each test spindle tested two bearings and was driven by a 10-horsepower alternating-current electric motor. Shaft speed was essentially constant and was reached soon after startup.

Version 1 of the test rig (Figure 28) shows a view of the test spindle as initially set up; two spindles were fabricated, designated A and B, respectively, in this report. A plain housing was installed on the drive end of the rig, and a grease-circulating device was installed on the outboard end. After the first test on each spindle, the grease circulating devices were changed from the "uphill" side of the outboard housing to the "downhill" side, as shown in Figure 29 (test rig, Version 2). At the same time, labyrinth seals with 0.010-inch diametral clearance were installed on both sides of each housing.

Three tests ran with no grease circulating or flinging device in either housing, as shown in Figure 30 (test rig, Version 3). Final testing was done with washer-type grease flingers on both sides of each test bearing to retard grease migration, as shown in Figure 31 (test rig, Version 4).

During the initial setup of the first spindle, it was found possible to increase misalignment by belt tension. Therefore, a procedure was established whereby the belt was tightened only until it began to change alignment, as shown by a 0.001-inch dial indicator. However, bearing torque had a tendency to tighten the belt and change misalignment during dynamic operation. During testing, generally greater cage wear from misalignment was observed on the drive-end bearing than on the outboard bearing.

Figures 32 through 41 show the following test items: Figure 32 is a photograph of the two test rigs as originally set up. Thermal insulation and heat lamps, to maintain 180°F ambient temperature, are not shown. Figure 33 is a photograph of the drive end of a spindle after initial test. Figure 34 is a photograph of the outboard end of a spindle after initial test. Figure 35 is a photograph of a partially disassembled drive end housing after the initial test. Figure 36 shows the outboard housing of spindle A, in Version 2 configuration, after the second test.

Figures 37 through 41 are photographs of rig components or partial assemblies. Figure 37 shows a bearing with an inner land riding S-Monel cage installed in a housing. Figure 38 shows rotating and stationary components of the grease circulating device used in Versions 1 and 2. Figure 39 shows the bearing and housing of Figure 37, with the grease circulating device installed. Figure 40 shows a labyrinth seal, used in rig Versions 2, 3, and 4, attached to a housing end plate.

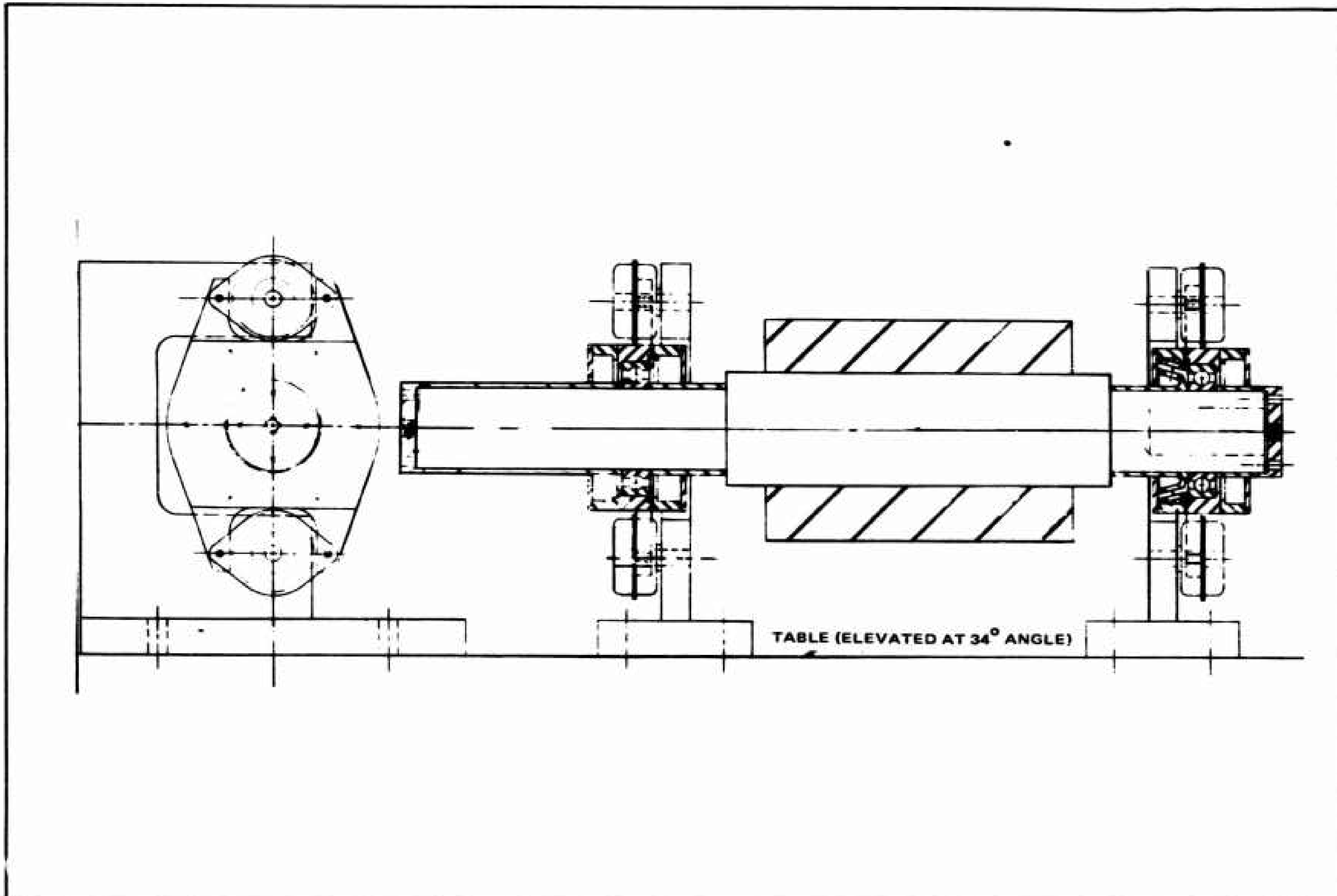


Figure 28. Test Rig, Version 1.

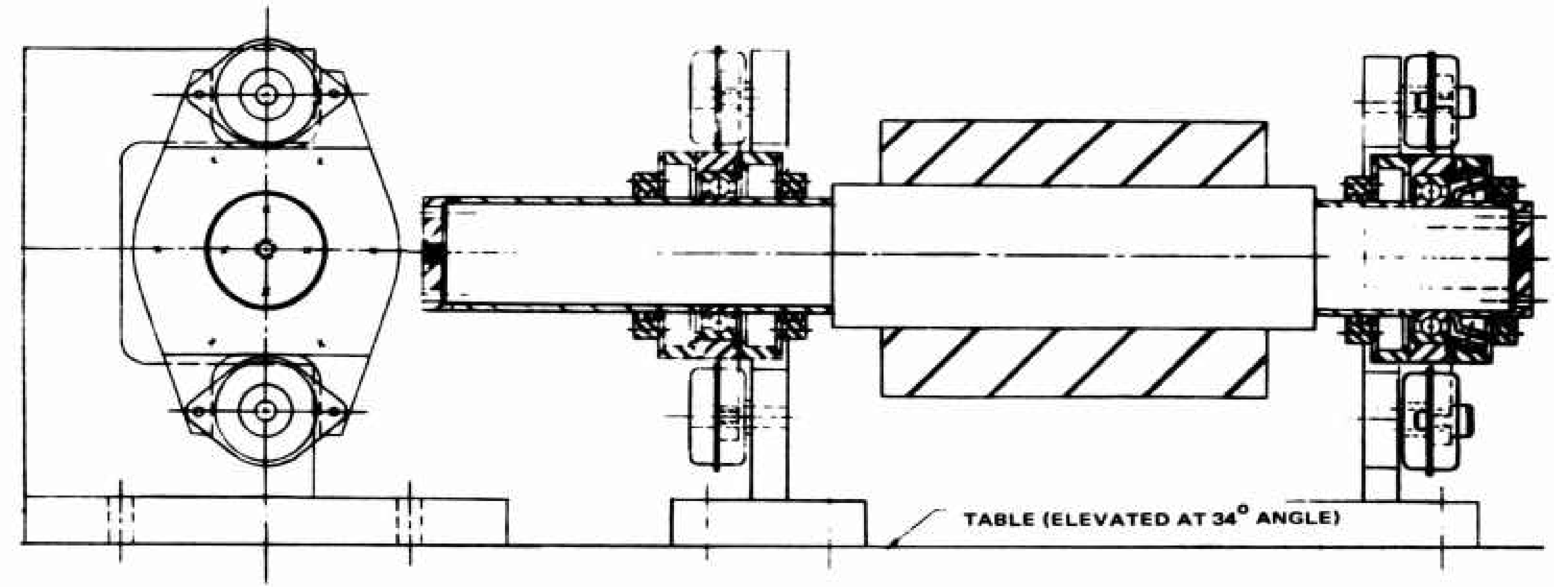


Figure 29. Test Rig, Version 2.

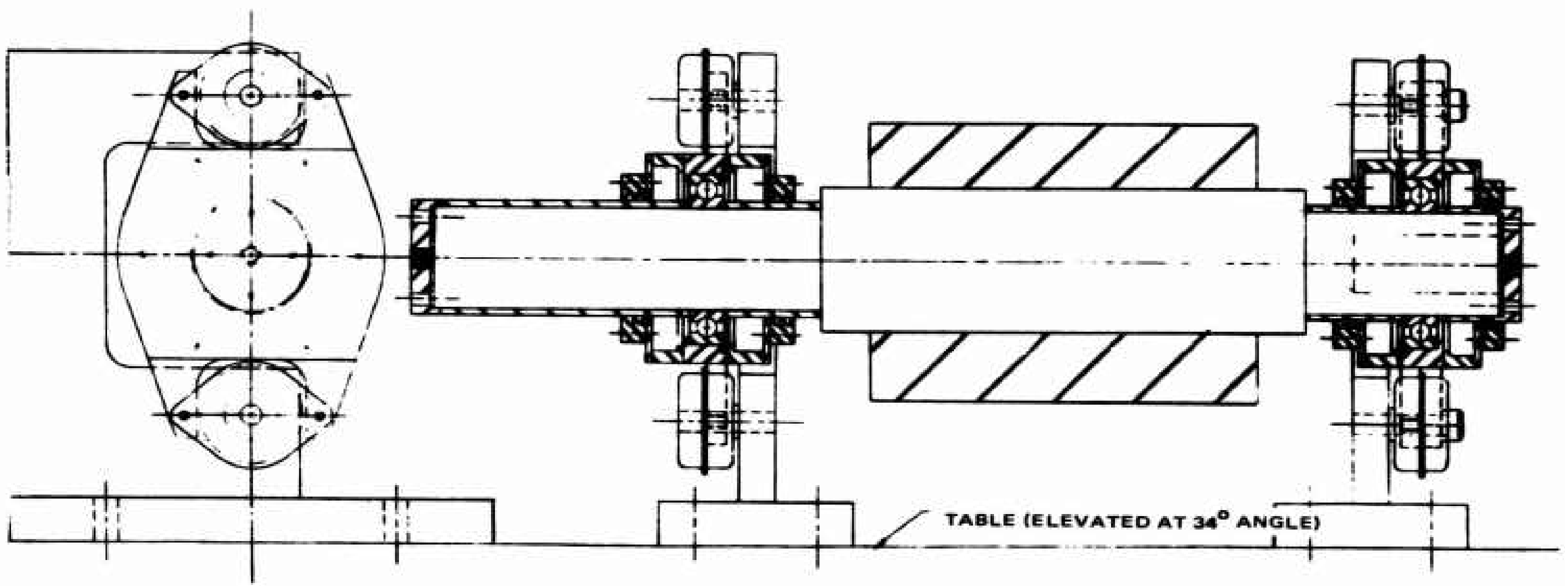


Figure 30. Test Rig, Version 3.

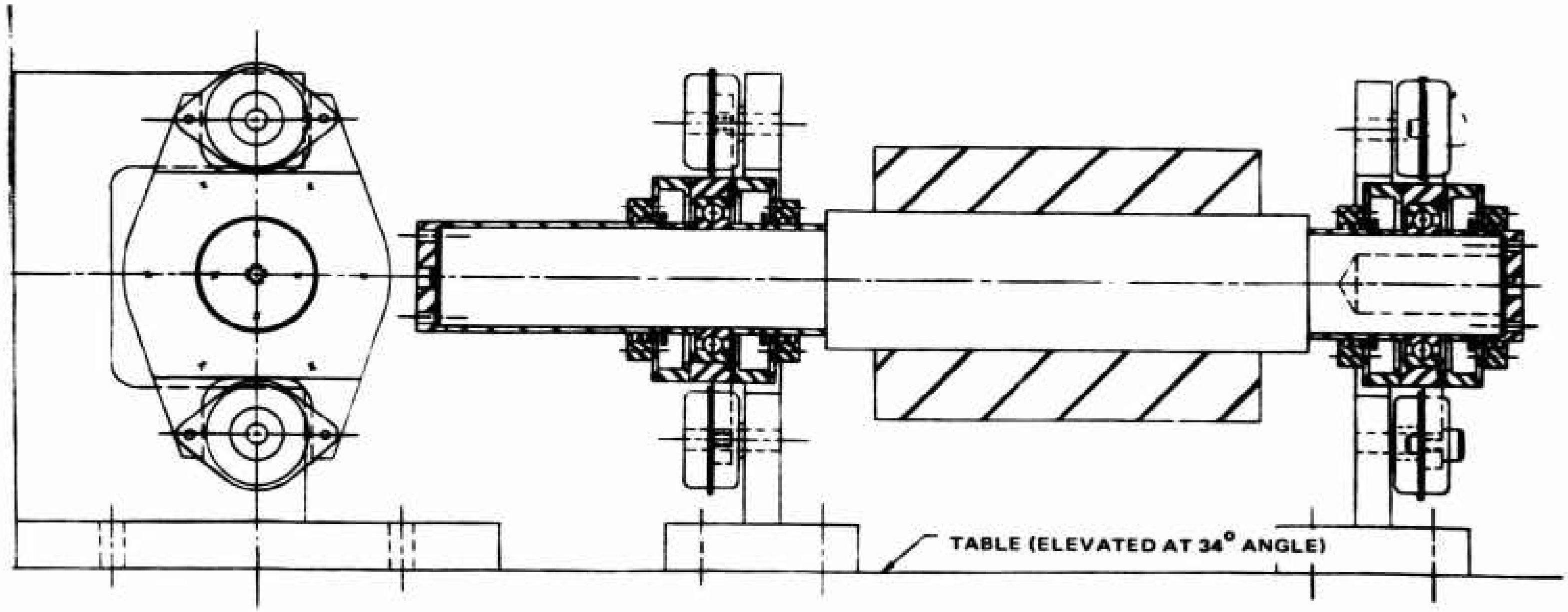


Figure 31. Test Rig, Version 4.

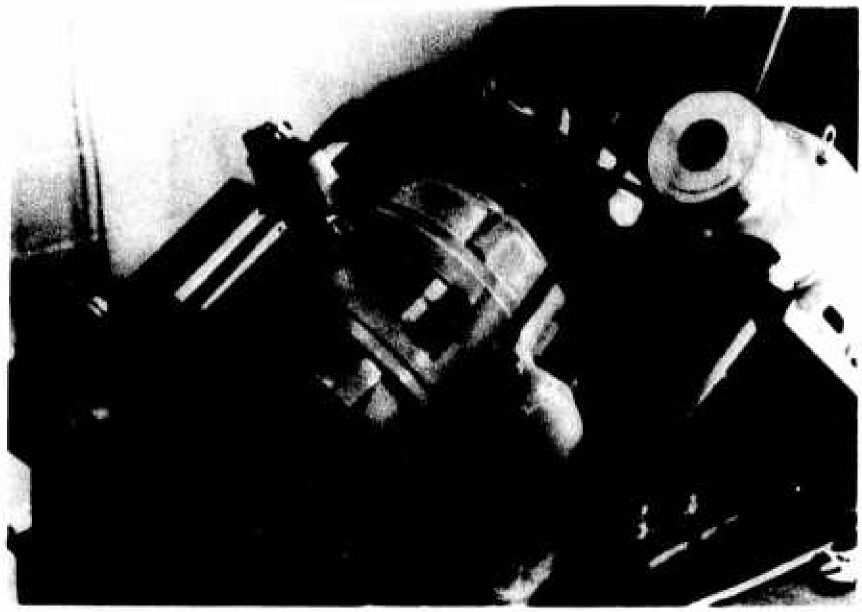


Figure 32. Basic Test Machine Setup.

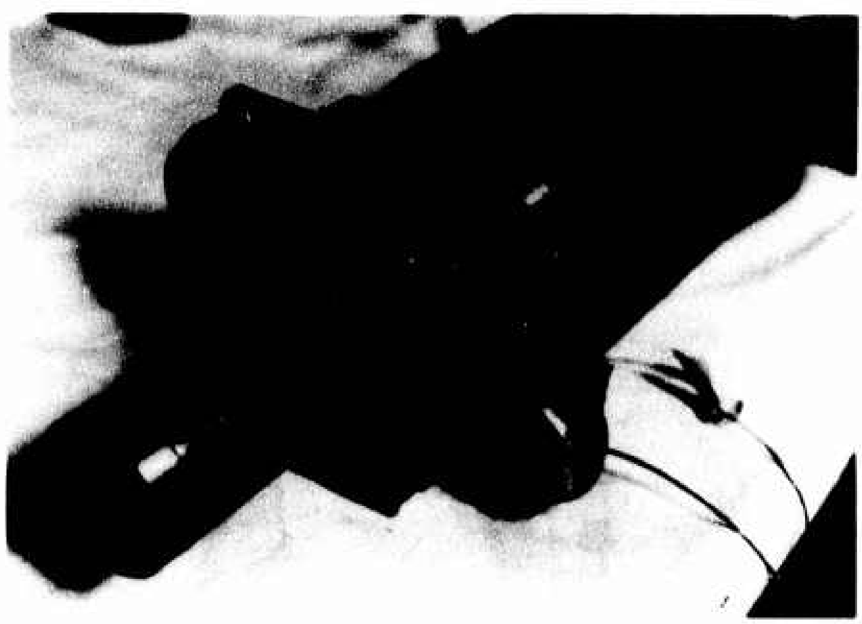


Figure 33. Drive End, Version 1.

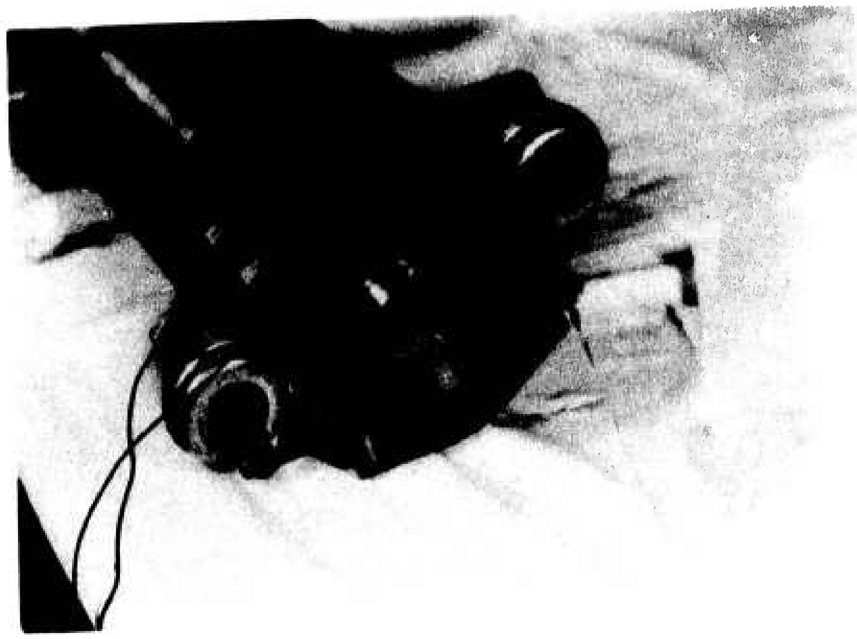


Figure 34. Outboard End, Version 1.

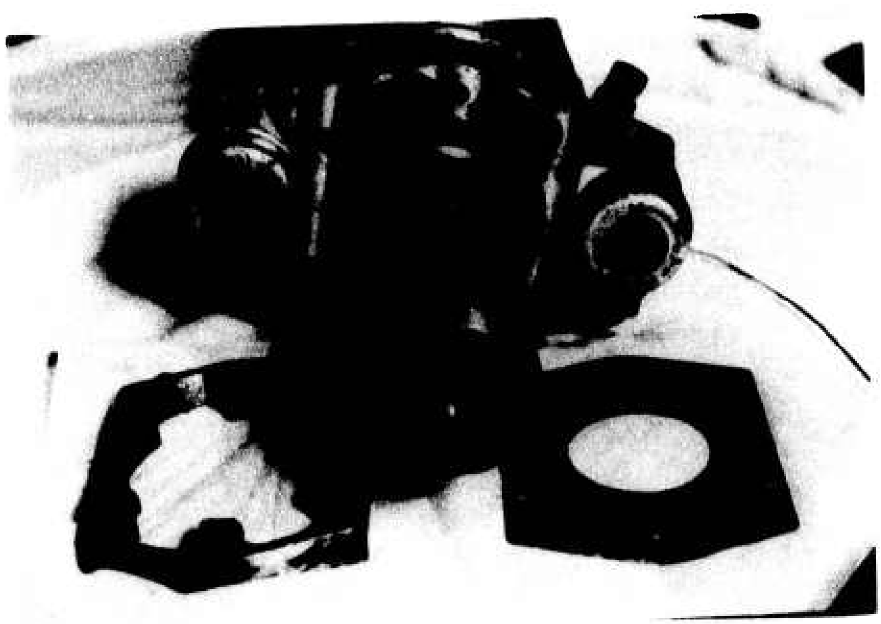


Figure 35. Partly Disassembled Housing.

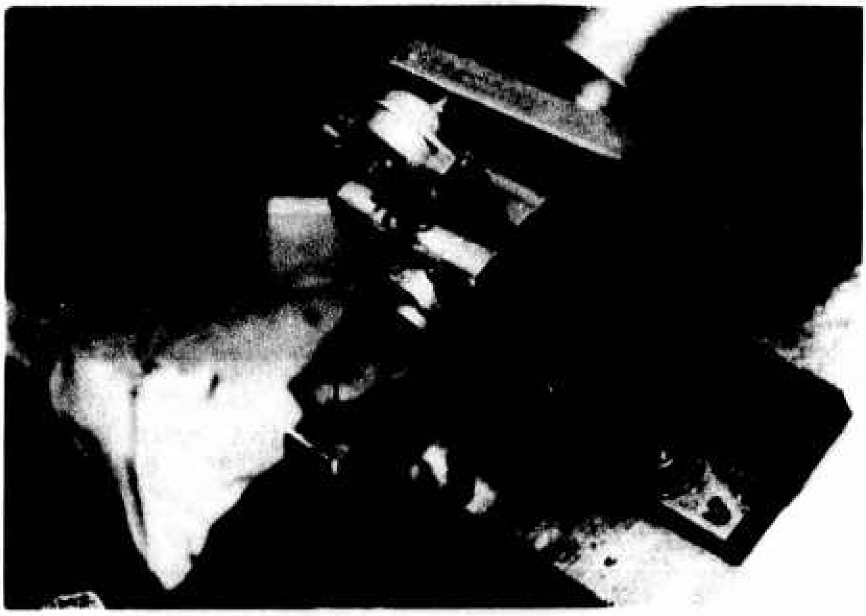


Figure 36. Outboard End, Version 2.

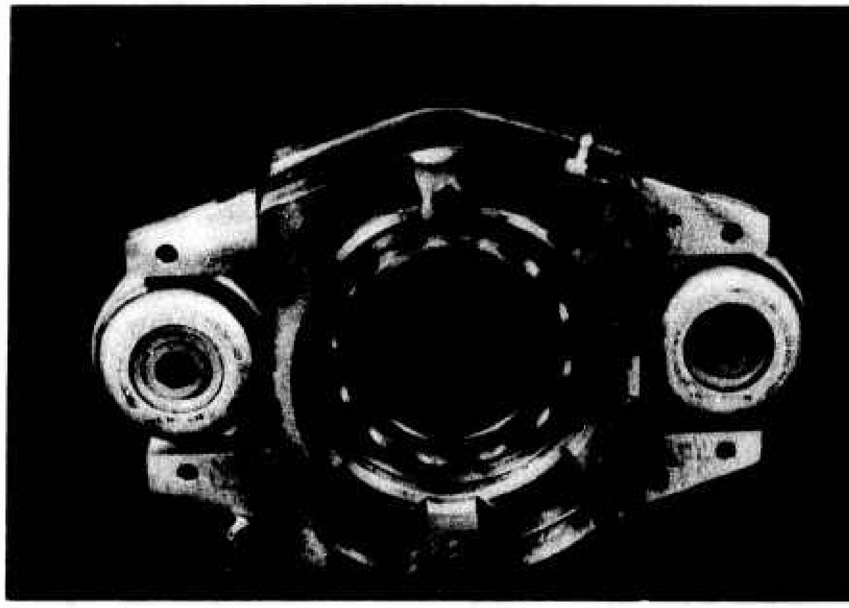


Figure 37. Housing With Bearing (Without End Plates).

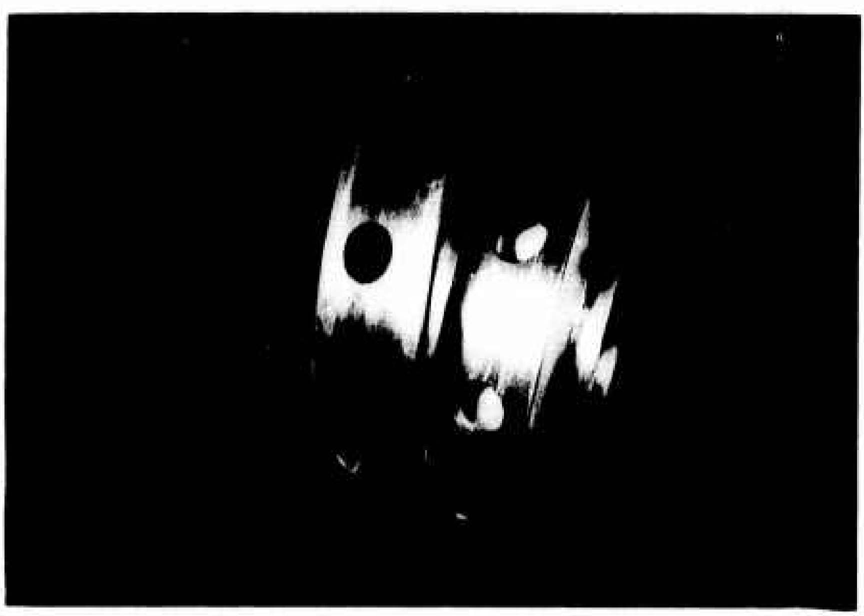


Figure 38. Grease Circulating Device.

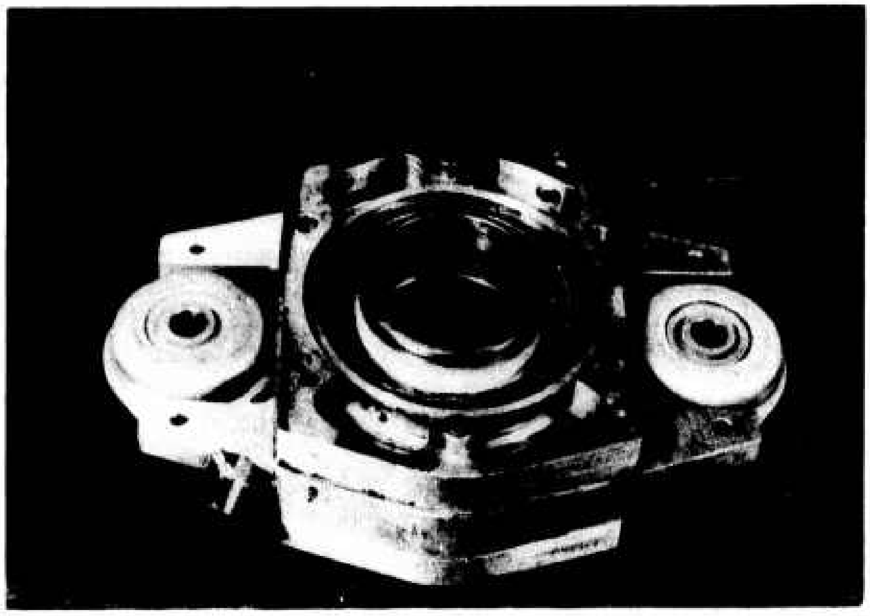


Figure 39. Housing With Grease Circulating Device Installed.

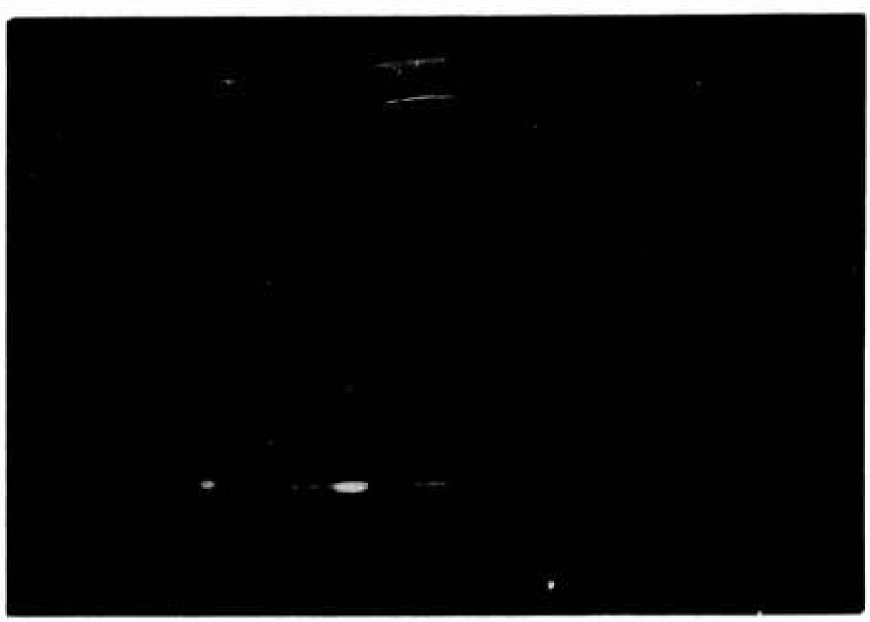


Figure 40. End Plate With Labyrinth Seal.

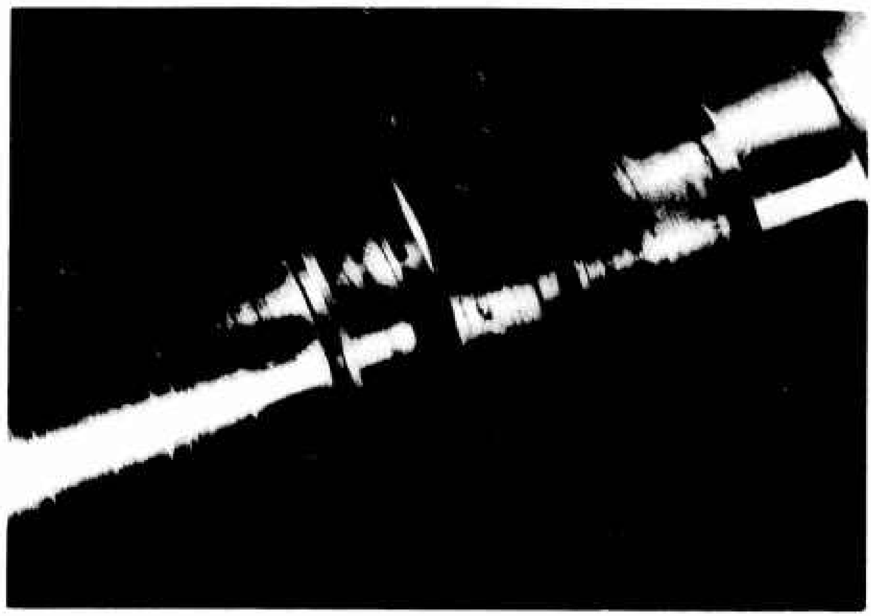


Figure 41. Bearing and Flingers (Version 4) on Shaft.

Figure 41 shows the drive end of a shaft with a bearing and spacers, and the grease flingers of Version 4 installed.

At the conclusion of the first series of bearing tests, it was observed that certain aspects of the investigation needed further work. Cage wear and cage breakage, which appeared to be related to misalignment, had occurred. Grease purging had been reduced but not eliminated by shaft flingers and labyrinth seals. As the bearings were modified to accommodate greater misalignment, additional problems of grease sealing were experienced because of the labyrinth seal rubbing the shaft. Therefore, the diametral clearance of the labyrinth seal was increased to 0.020 inch maximum. Also, the outer diameters of the shaft flingers were increased from 2.9 to 3.5 inches to help compensate for the increased clearance of the seals.

Prior to the final series of tests, Version 4 of the test rig (Figure 31) was again modified to more closely simulate the actual helicopter shaft support housing. The final modifications were as follows:

1. In prior testing, each shaft was belt driven by a motor. There was a possibility that belt tension increased misalignment and that belt slap added to vibration. Therefore, the arrangement was changed to have the motors drive separate pillow-block mounted shafts which were flexibly coupled to the test shafts.

2. The elastomeric mounts were previously bolted to stanchions. In accordance with Boeing Vertol practice, the bolts were replaced with studs onto which the elastomeric mounts could slide. The weight of the rig acting at the 34-degree angle caused the shaft to position itself axially, and all axial load tended to be taken by one bearing.

The test rig was checked out by a 91-hour run on two slave bearings to ensure that the above modifications were acceptable. Each test was started without external heat to determine self-generated temperature. After stabilization, normally within an hour, two 250-watt infrared lamps were turned on each housing and the rig was covered with an insulated box. Previous experience indicated that these heat lamps, plus bearing heat, were sufficient to achieve the 180°F ambient temperature. It was sometimes necessary to turn off one or more lamps, depending on bearing heat development.

INSTRUMENTATION

Test rig speeds were measured by a calibrated strobe-type tachometer. No indication of speed change or belt slippage was observed throughout testing.

Outer race and external housing temperatures were measured by chromel-alumel thermocouples and recorded on Leeds and Northrup multiple point recorders.

Vibration, in two planes, was measured on each housing by Columbia accelerometers and indicated on a Panoramic Sonic Analyzer. These vibration indications were recorded periodically by a Polaroid Land Camera.

Ambient temperature was maintained by a combination of insulation and infrared lamps. At times, self-generated heat was sufficient to maintain the 180°F scheduled ambient temperature; at other times, considerable external heat was required.

Spindle unbalance was obtained by first balancing the shaft on a Gisholt dynamic balancer, then removing a calculated amount of stock from the O.D. of the central portion of the shaft.

Initial misalignment was obtained by assembling the spindles with no misalignment, spiking the outboard support stanchion, then sliding the drive-end stanchion through a 15-minute arc toward the motor. Therefore, any effect of belt tension was to increase misalignment rather than relieve it.

TEST DATA AND RESULTS

GREASE EVALUATION TESTS

Before full-scale bearing tests were conducted, a preliminary grease evaluation test program was conducted. This testing was done on the MRC 1000°F Grease Test Spindle as shown in Figure 26. The test spindle was set up with two identical MRC 204-S-17 bearings which were prepacked with the test grease. The bearing on the pulley end of the shaft was called the "front" bearing, and the other bearing was referred to as the "rear" bearing. Each bearing was packed 30% full with grease.

Only the outer race temperature of the rear bearing was recorded during each test. Therefore, failure or deterioration of the front bearing was not indicated by temperature data.

All tests were conducted under the following conditions:

Speed	32,000 rpm
Load	50 pounds axial
Lubrication	Grease, as specified, prepacked
Temperature	No external heat
Test Duration	300 hours or failure

Five greases were evaluated. Table II provides a brief description of the greases tested and also shows the quantity of grease packed into the bearing prior to testing.

Table III provides a summary of the grease evaluation tests. All bearings except those used in test 14 had approximately 0.0010 inch unmounted radial clearance. Bearings used in test 14 had the radial clearance increased to approximately 0.0020 inch.

A review of the test data showed that Aeroshell 22 (MIL-G-81322) grease provided the best test results. Therefore, this grease was selected as the lubricant for all full-scale (MRC 111-KS) ball bearing tests.

As noted in Table II, two manufacturers supplied grease conforming to MIL-G-81322 grease. Aeroshell 22 provided much better performance than Mobilgrease 28. Several investigations were conducted to determine the large variation in performance of these two greases. Mobil Oil Company supplied a different batch of Mobilgrease 28 which permitted the MRC 204-S-17 bearings to operate for 300 hours under the above conditions. The only possible reason established for the differences was a variation in early batches of the Mobil

TABLE II. SUMMARY DESCRIPTION OF GREASES TESTED

Grease Trade Name	ASU-31052	Mobilgrease 24	Mobilgrease 28	Aeroshell 22 (XSG-6377)	EG-545
Manufacturer	American Oil Co.	Mobil Oil Co.	Mobil Oil Co.	Shell Oil Co.	CATO Oil and Grease Co.
Thickener	Arylurea	Non-Soap, Organic	Bentone	Microgel (Clay)	Non-Soap
Oil	Silicon Fluid	Silicon Fluid	Synthetic Hydrocarbon (150 sus at 100°F)	Synthetic Hydrocarbon	Silicon fluid & Polyphenylether
Applicable Temperature Range	-100°F to +450°F	-100°F to +450°F	-65°F to +350°F	-65°F to +350°F	-20°F to +600°F
Military Specification	MIL-G-25013D	MIL-G-25013D	MIL-G-81322	MIL-G-81322	None
Weight of Grease per Bearing*	3.3 Grams	3.4 Grams	3.4 Grams	2.8 Grams	3.8 Grams

*All bearings were prepacked 30% full of grease.

TABLE III. SUMMARY OF MRC HIGH-TEMPERATURE GREASE TEST SPINDLE RESULTS (Rear Bearing Only)

Test No.	Grease	Hours	Running Temp (°F)	Maximum Temp (°F)	Remarks
1	ASU-31052	3.0	250	250	Broken cage in front bearing; rear bearing fairly good
2	ASU-31052	0.7	-	250	High torque and squealing; both bearings looked fairly good after test
3	XSG-6377 (Aeroshell 22)	201.2	192	240	Rear bearing rather dry; front bearing OK
4	Mobil 24	1.2	230	270	High torque; both bearings looked fairly good after test
5	Mobil 28	1.3	192	278	Broken cage in rear bearing; front bearing fairly good after test
6 thru 11 (Test rig trouble. Test spindle vibration problem due to spindle unbalance.)					
12	XSG-6377	305.6	170	185	New spindle used. No failure, 0.35 gram grease in each bearing after test
13	ASU-31052	2.0	-	185	Front bearing overheated; rear bearing OK
14	ASU-31052	0.5	-	190	Broken cage in front bearing; grease gone; rear bearing OK
15	Mobil 24	0.9	-	300	Front bearing slightly discolored; rear bearing OK
16	Mobil 28	35.6	220	253	Rear bearing was black and rather dry; front bearing OK
17	EG-545	29.3	220	235	Broken cage in front bearing; rear bearing fairly good
18	XSG-6377	331.8	180	202	No failure, 0.30 gram grease in each bearing after test

grease. Present test data indicate that greases conforming to MIL-G-81322 should provide the best lubricant for the HLH engine shaft support bearing.

FULL-SCALE RIG TESTING

A total of 18 full-scale rig tests were conducted during this program; a total of 32 bearings were tested. Several bearings were reworked and used several times. A summary of the various bearing and housing configurations with a brief description of the test results is shown in Table IV. An additional description of each test will be provided to supplement this table. Table V provides a brief summary of bearing operating temperatures for the various cage configurations used in tests 1 through 15.

Two test machines, A and B, were used to conduct the full-scale testing. Testing was initially conducted using two basic housing designs and four bearing configurations. The test results from the first series of tests were analyzed, and additional modifications were incorporated into both the bearing and housing in order to achieve the desired goal. After test 12, the housing configuration was basically held constant and the bearing was modified to optimize the final bearing assembly configuration.

The results of the full-scale bearing tests, and a description of pertinent data for each of the tests, are presented in the following paragraphs.

Test 1

This test evaluated bearings S/N E9 and E2 with standard pressed steel cages and the grease circulating device in a housing (Figure 42a) versus a standard bearing housing (Figure 42b). Each housing was packed with 7 ounces of grease. After 71.7 hours, temperature of the drive-end bearing indicated inadequate lubrication. Six ounces of grease were added to the drive-end housing and 3 ounces to the outboard. At 171.7 hours, 6 more ounces of grease were added to the drive-end housing. The test was suspended as scheduled at 300 hours. Both bearings were serviceable, but bearing S/N E9, the drive-end bearing, indicated three overheated ball pockets. See Figures 43 and 44 for examples of the bearings with standard pressed steel cages after test. Figure 45 shows the inside of both housings after test. The housing with the grease circulating device appeared to retain the grease better than the standard housing.

TABLE IV. SUMMARY OF FULL-SCALE BALL BEARING TEST RESULTS

Test No.	Mach.	Rig Config.	Bearing S/N and Cage Description: Drive End	Bearing S/N and Cage Description: Outboard End	Test Hours	Remarks
1	A	1	E9 (Standard Pressed Steel)	E2 (Standard Pressed Steel)	300.0	E9 indicated three overheated ball pockets.
2	B	1	E4 (Inner Riding Monel, Design 1)	E17 (Inner Riding Monel, Design 1)	300.0	Both cages wore on bore. E17 lost 5 rivets with 2 additional loose.
3	A	2	E13 (SP-21 Insert Type)	E16 (SP-21 Insert Type)	27.7	Erratic behavior.
4	B	2	E14 (Design 2 Inner Riding Monel)	E26 (Design 2 Inner Riding Monel)	300.0	Both cages wore on bore. E14 had heaviest wear.
5	A	2	E1 (Teflon Coated, Pressed Steel)	E11 (Teflon Coated, Pressed Steel)	246.2	Cage separated on E1.
6	A	2	E22 (Outer Riding Monel)	E21 (Outer Riding Monel)	71.1	Lube failure in E21.
7	A	3	E22 (Outer Riding Monel)	E19 (Outer Riding Monel)	22.1	No failure. Test suspended to permit change of rig configuration.
8	A	4	E22 (Outer Riding Monel)	E19 (Outer Riding Monel)	53.8	Cage in E22 broke.
9	B	2	E18 (Reworked Insert Type)	E24 (Reworked Insert Type)	5.2	E24 broke steel frame.
10	B	3	E18 (Reworked Insert Type)	E3 (Teflon Coated Pressed Steel)	1.5	Grease circulating flinger removed prior to test. E18 broke steel frame.

TABLE IV - Continued

Test No.	Mach.	Rig Config.	Bearing S/N and Cage Description				Test Hours	Remarks
			Drive End		Outboard End			
11	B	3	E7	(Standard Pressed Steel)	E3	(Teflon Coated Pressed Steel)	110.0	E7 broke cage.
12	A	4	E15	(Design 4 Inner Riding Monel-Teflon Coat on Inner)	E20	(Design 3 Inner Riding Monel-Teflon Coat on Inner)	300.0	Wear on bore of both cages. Teflon coat abraded.
13	B	4	E23	(Silver-Plated Steel-Teflon Coat on Inner)	E6	(Silver-Plated Steel-Teflon Coat on Inner)	300.0	Both cages wore. E23 cage broke. Teflon coat abraded.
14	A	4	31	(Design 3, Inner Riding Monel-Teflon-S Coat on Inner-Race Curvatures)	32	(Design 3, Inner Riding Monel, Increased Race Curvatures)	302.6	Both cages wore. 31 cage separated. Teflon coat abraded.
15	A	4	2	(Design 5, Inner Riding Monel-Sermetel 72 Coat on Inner)	2	(Design 5, Inner Riding Monel-Sermetel 72 Coat on Inner)	167.0	Both cages wore. 2 cage separated. Sermetel 72 coat abraded.
16	A	4	3	(Design 6, Ball Riding Monel)	4	(Design 6, Ball Riding Monel)	300.0	Both bearings in good condition.
17	A	4	5	(Design 6, Ball Riding Monel)	4	(Design 6, Ball Riding Monel)	300.0	Bearing 5 in excellent condition. Bearing 4 showed ball and cage wear.
18	B	4	6	(Design 6, Ball Riding Monel)	1	(Design 6, Ball Riding Monel)	300.0	Both bearings in excellent condition.

TABLE V. SUMMARY OF BEARING TEMPERATURES FOR TESTS 1 THROUGH 15			
Bearing No.	Cage Type	Outer Race Temperature	
		Unheated (°F)	Heated (°F)
E-2	Pressed Steel	130 (on a restart)	180-190
E-9	Pressed Steel	150 (on a restart)	180-200
E-7	Pressed Steel	150	165-180
E-11	Pressed Steel, Teflon Coated	146	180
E-1	Pressed Steel, Teflon Coated	155	180
E-3	Pressed Steel, Teflon Coated	160	150-165
E-16	SP21 Inserts	165-210	Not heated
E-13	SP21 Inserts	160-185	Not heated
E-18	SP21 Inserts (Reworked Design)	155 at start	Not heated
E-24	SP21 Inserts (Reworked Design)	172 at start	Not heated
E-22	Outer Riding S-Monel	190-200	190
E-19	Outer Riding S-Monel	170	180, dropping to 155
E-21	Outer Riding S-Monel	220	Not heated
E-23	Silver-Plated Steel	145	150-180
E-6	Silver-Plated Steel	Erratic, as low as 130, mostly over 180	180-200
E-4	Design 1, S-Monel	170	180
E-17	Design 1, S-Monel	145	180
E-26	Design 2, S-Monel	165, dropping to 130	180-190
E-14	Design 2, S-Monel	120	180-190
E-20	Design 3, S-Monel	130	180
E-15	Design 4, S-Monel	135	180-200
E-32	Design 3, S-Monel (Incr. Curvatures)	110	170
E-31	Design 3, S-Monel (Incr. Curvatures)	102	170
1	Design 5, S-Monel (Incr. Curvatures)	190, dropping to 170	170-180
2	Design 5, S-Monel (Incr. Curvatures)	150, dropping to 115	160-180
E-12	Pressed Steel, Teflon Coated	130, dropping to 100	Not heated*
E-10	Pressed Steel, Teflon Coated	130	Not heated*

*Slave bearing rig checkout test.

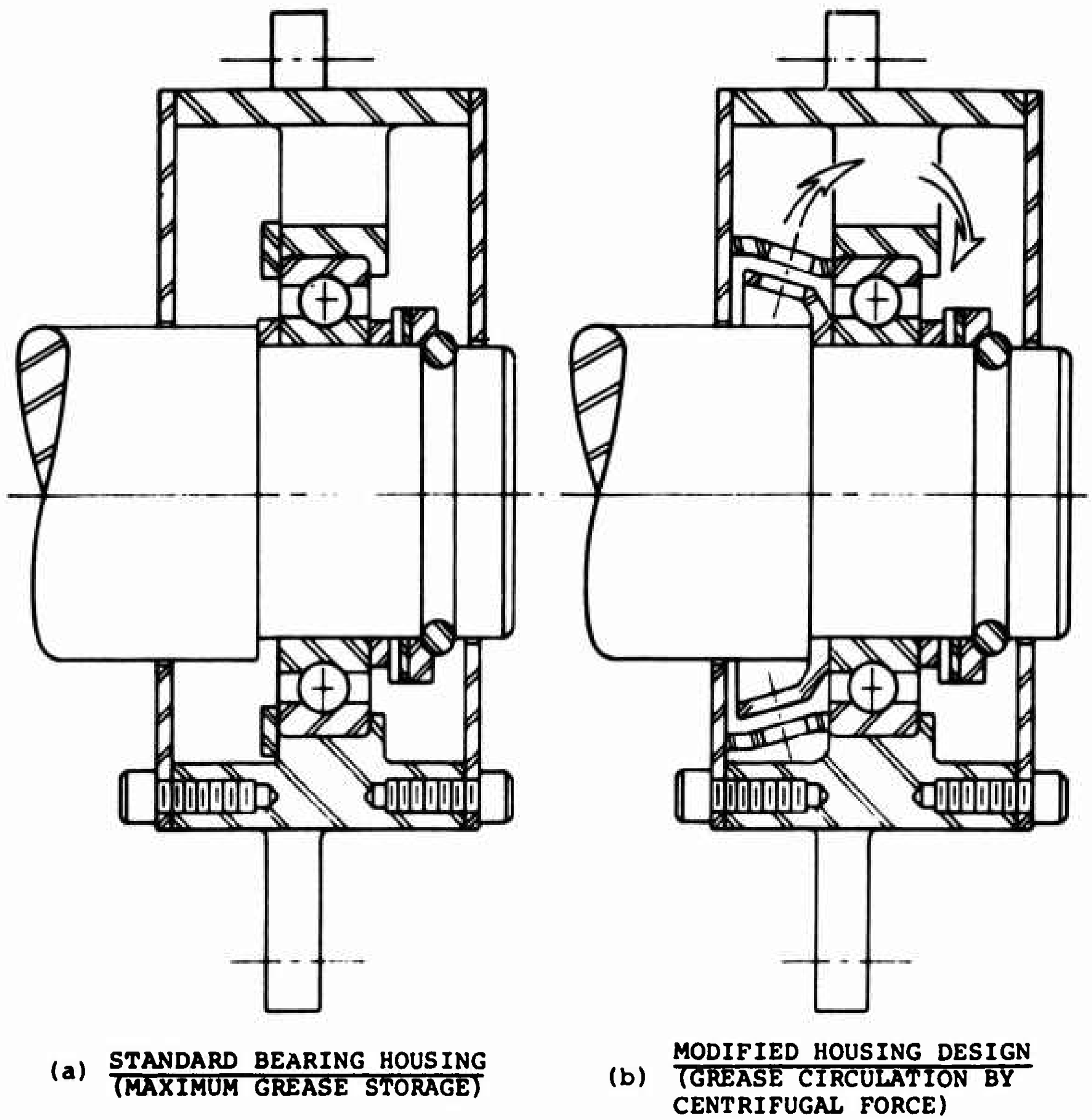


Figure 42. Standard and Modified Bearing Housing Design.

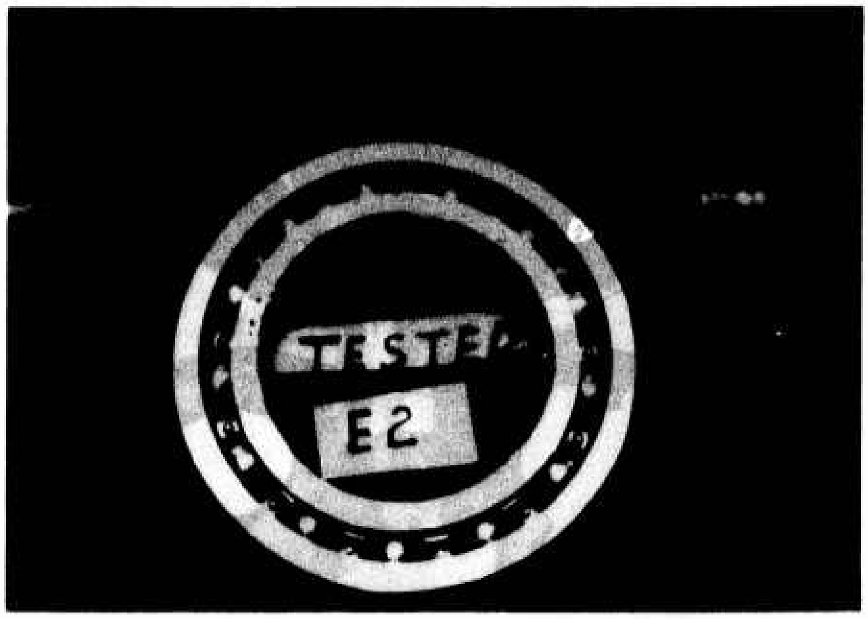
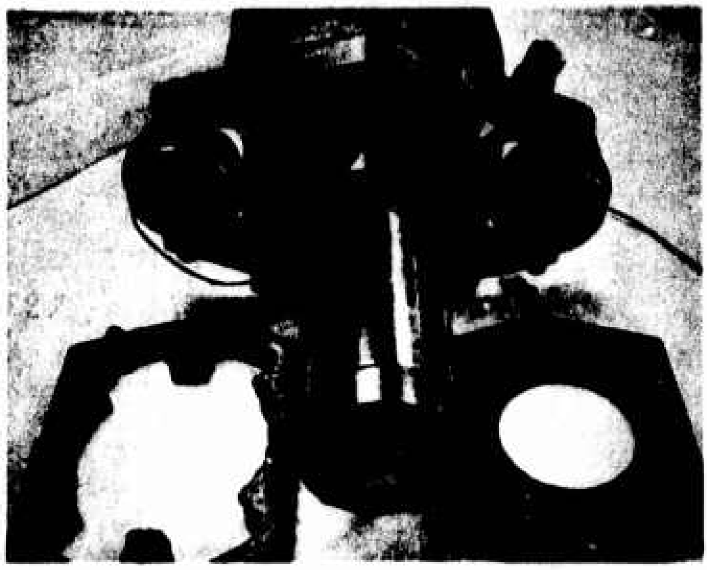
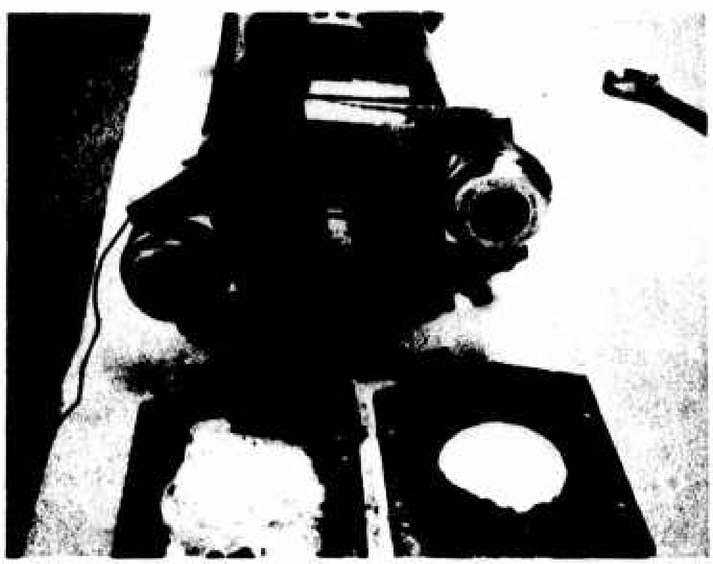


Figure 43. Bearing S/N E2, After Test 1.

Figure 44. Bearing S/N E9, After Test 1.



STANDARD HOUSING AFTER COMPLETION OF 300-HOUR TEST
(57 HOURS AFTER LAST REGREASING)
(NOTE LACK OF GREASE COMPARED TO MODIFIED HOUSING)



MODIFIED HOUSING WITH GREASE CIRCULATING DEVICE
AFTER COMPLETION OF 300-HOUR TEST
(228 HOURS AFTER LAST REGREASING)

Figure 45. Bearing Housing After Test 1.

Test 2

This test evaluated bearings S/N E4 and E17 with the original (Design 1) inner land riding S-Monel cage and the grease circulating device in a housing. Both housings were initially packed with 7 ounces of grease. At 178.0 hours and 278.8 hours, it was necessary to regrease the drive-end housing, using 6-1/2 ounces each time. No regreasing was necessary in the outboard housing, and it had 5 ounces left at the end of 300 hours. The bore of the cage of the drive-end bearing had worn to a maximum depth of 0.040 inch. The cage in the outboard bearing had worn relatively uniformly 0.010 to 0.014 inch on the diameter, but it lost 5 rivets and 2 others were loosened.

Note: Tests 1 and 2

The first two tests indicated that neither cage design was entirely satisfactory. A decision was made to modify the inner land riding cage design and to try an outer land riding design.

The housings with grease circulating devices had purged considerably less grease than had the plain housings (Figure 46); at the same time, grease in the area of the devices appeared heavily worked. Consequently, a decision was made to install the devices on the "downhill" side of the outboard bearing and to install labyrinth seals on all housings. See Figures 47, 48, and 49 for a picture of a bearing before test and pictures of the bearings after test.

Test 3

This test evaluated the insert type cage (fail-safe concept) in bearings S/N E13 and E16.

Erratic performance of bearings resulted in several teardowns and inspections, so housing and seal evaluations were impossible. Basically, the bearings had difficulty getting enough grease and were partially starved. Their torque tended to be high or to vary, causing the shaft to change its misalignment, and causing the close-running labyrinth seals to contact the shaft. The test was terminated prematurely, and bearings were inspected. Balls and races were cloudy, indicating surface deterioration. Figures 50 and 51 show one of these bearings before and after test.

Test 4

This test evaluated bearings S/N E14 and E26 with Design 2 of the inner land riding S-Monel cage, plus housing design modifications. Both housings were packed with 7 ounces of grease

PURGED GREASE FROM HOUSING WITH
GREASE CIRCULATING DEVICE

PURGED GREASE FROM STANDARD HOUSING

Figure 46. Purged Grease From Housings After Test.

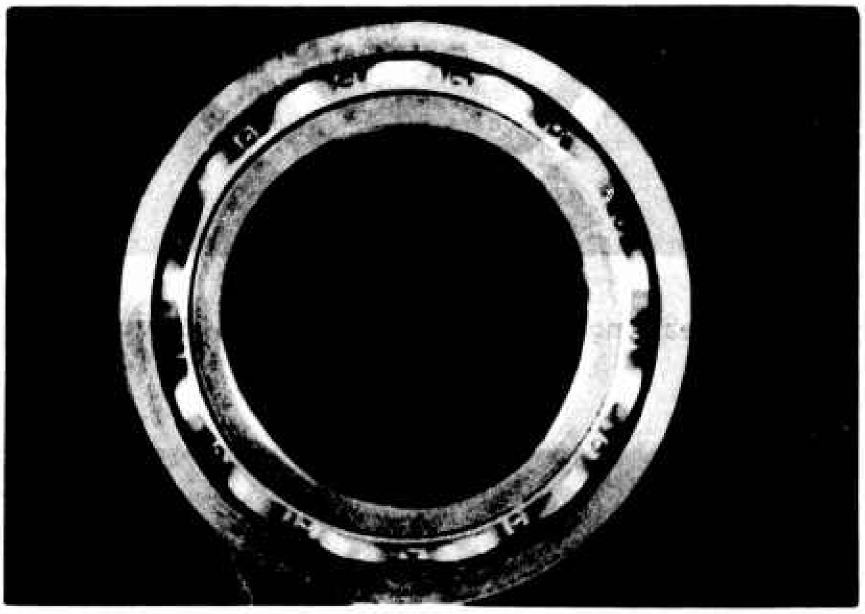


Figure 47. Typical Specimen Before Test (Inner-Land-Riding S-Monel Cage, Design 1).

Figure 48. Bearing S/N E17 After Test, Several Rivets Lost.

Figure 49. Bearing S/N E4 After Test, Grease Still in Bearing.

Figure 50. Bearing S/N E13 Before Test (SP-21 Insert Design).

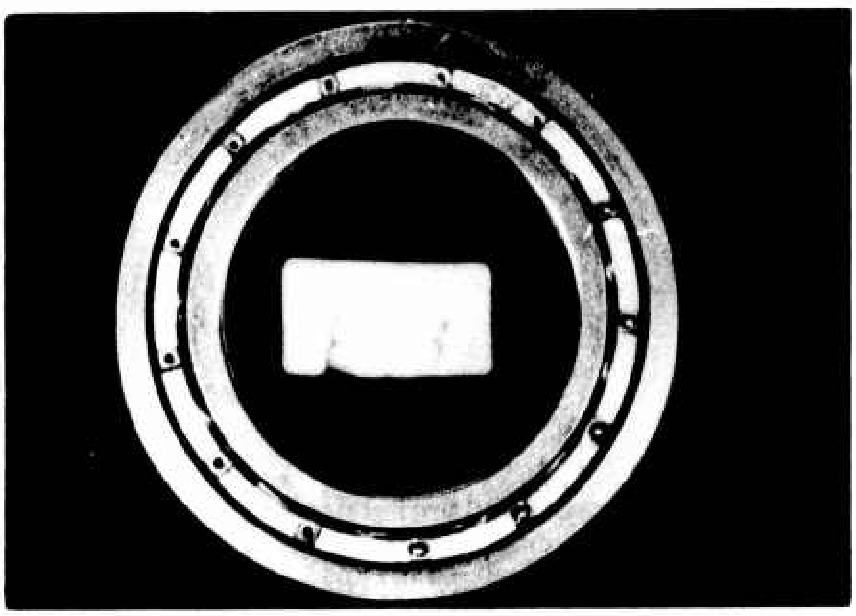


Figure 51. Bearing E13 After Test (Some Rivets Have Been Removed for Disassembly).

at the start. Grease additions were made as follows:

Time (hours)	Drive End (ounces)	Outboard End (ounces)
63.0	2	2
91.0	3	2
102.7	2	2
280.9	-	3

After 300 hours the test was suspended. Both bearings had worn on the bore, with the heaviest wear, approximately 0.035 inch, in the drive-end bearing. The outboard bearing had experienced maximum depth of wear of 0.018 inch. Neither cage had lost rivets. Figures 52, 53, and 54 show a bearing before test and the 2 bearings after test.

Housing design evaluations in this test were inconclusive.

Test 5

This test evaluated the Teflon-coated pressed steel cage, plus housing designs using bearings S/N E1 and E11. Grease additions were made as follows:

Time (hours)	Drive End (ounces)	Outboard End (ounces)
25.2	-	1
63.3	-	1
233.1	3	2
244.1	1	2

During the last several hours of operation, performance of the drive-end bearing was erratic, and the grease additions to this position were attempts to stabilize it. At 246.2 hours the rig was shut down and disassembled. The cage in the drive-end bearing was found to have lost all rivets and separated. The outboard end bearing was still intact. Figures 55, 56, and 57 show the bearing with Teflon-coated cage before test, and two bearings after test.

This test indicated a definite improvement in housing grease retention over rig configuration 1.

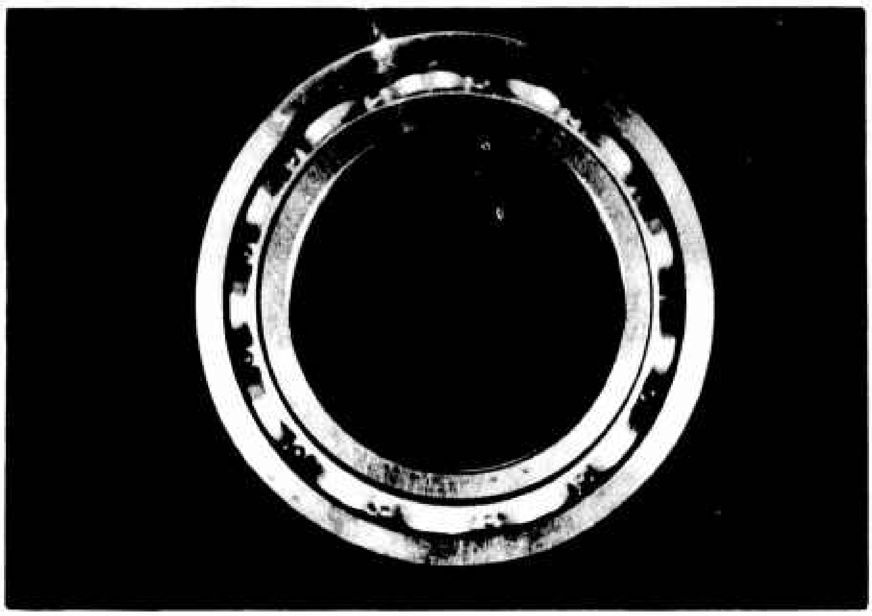


Figure 52. Bearing E14 Before Test (Cage Design 2).

Figure 53. Bearing E14 After Test.

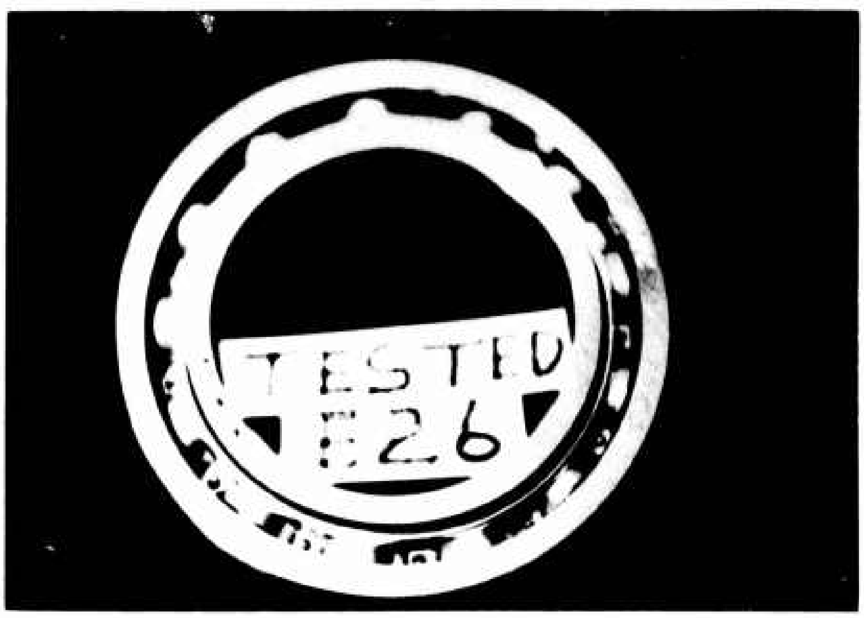


Figure 54. Bearing E26 After Test.

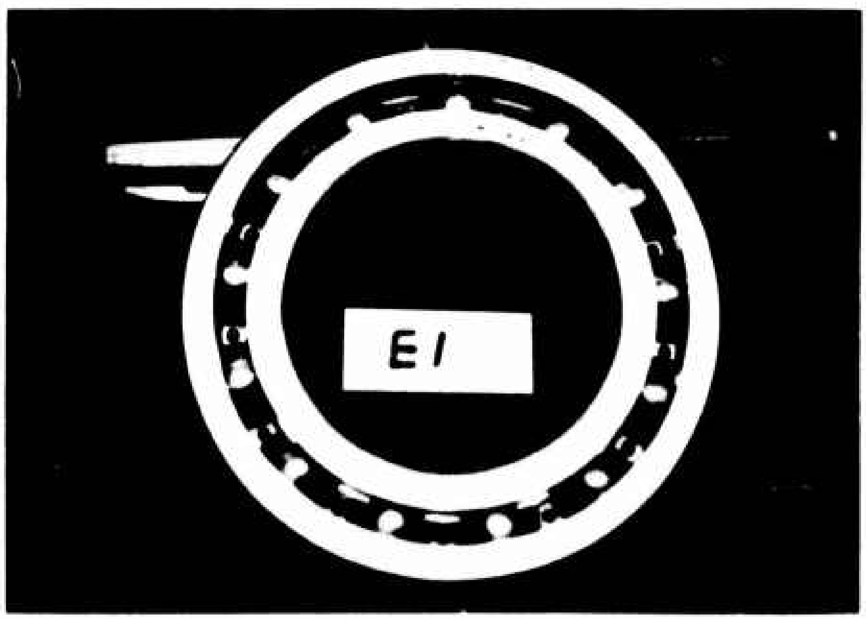


Figure 55. Bearing E1 Before Test (Teflon-Coated Pressed-Steel Cage).

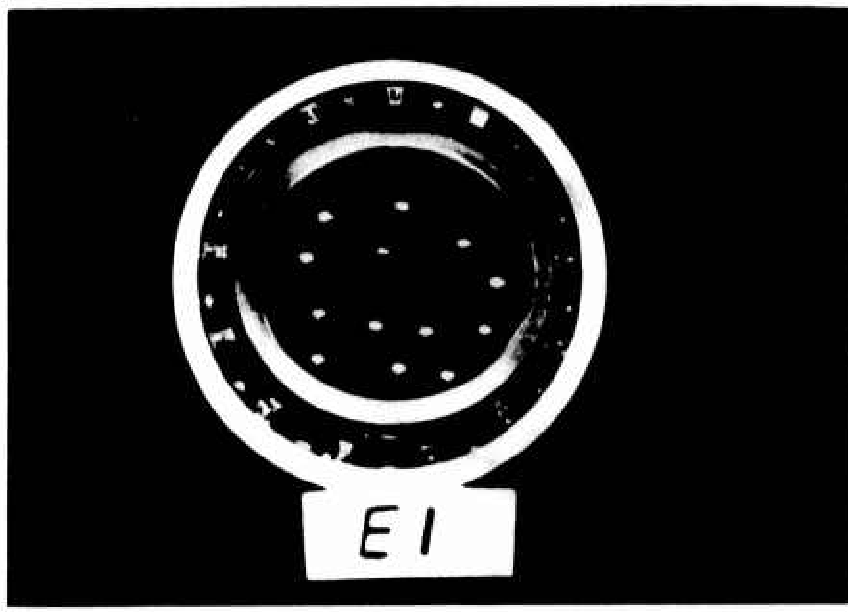


Figure 56. Bearing E1 After Failure in Test 5.

Figure 57. Bearing E11 After Test.

Tests 6, 7 and 8

These tests evaluated the outer land riding S-Monel design using bearings S/N E19, E21 and E22. Test 6 was stopped by a lubrication failure in the outboard bearing (S/N 21). The rig configuration was changed to No. 3 prior to test 7. Test 7 was stopped to again change the rig configuration to No. 4, and the same bearings were run in test 8. The drive-end bearing, E22, suffered a radial cage break. This bearing had run a total of 147 hours in the three tests. Figures 58, 59, and 60 show bearings with this cage design, before and after test.

This cage design proved to be inadequate and was dropped from further consideration. The lubrication failure in the outboard housing indicated that the grease circulating device, of test rig Versions 1 and 2, was not particularly effective and could be eliminated.

Tests 9 and 10

These tests evaluated bearings S/N E18 and E24 with reworked insert type cages (Figure 21) in which stock had been removed to facilitate grease lubrication. Figure 61 shows a bearing before test. All stock was removed from the steel frame. Both tests ended in very short time with broken cages (see Figures 62 and 63).

These tests indicated that we had removed too much stock in reworking the cages. However, removal of significantly less stock would not have provided needed space for grease. Consideration of these insert designs was dropped for this program.

Test 11

This test involved further consideration of pressed steel cages (ball riding, with and without Teflon coating using bearings S/N E7 and E3). After 110 hours, with no grease additions, the cage on the drive-end bearing (S/N E7) broke as shown in Figure 64. Performance, as measured by heat generation and vibration, was excellent until failure. This test again indicated that there is a reliability problem with pressed steel cages operating at high DN values.

Test 12

This test was a further evaluation of modified inner-land-riding S-Monel designs using bearings S/N E15 and E20. Both cages were silver-plated. The lands of both inner rings were coated with Teflon-S; the lands of the outboard bearing were further coated with FEP.

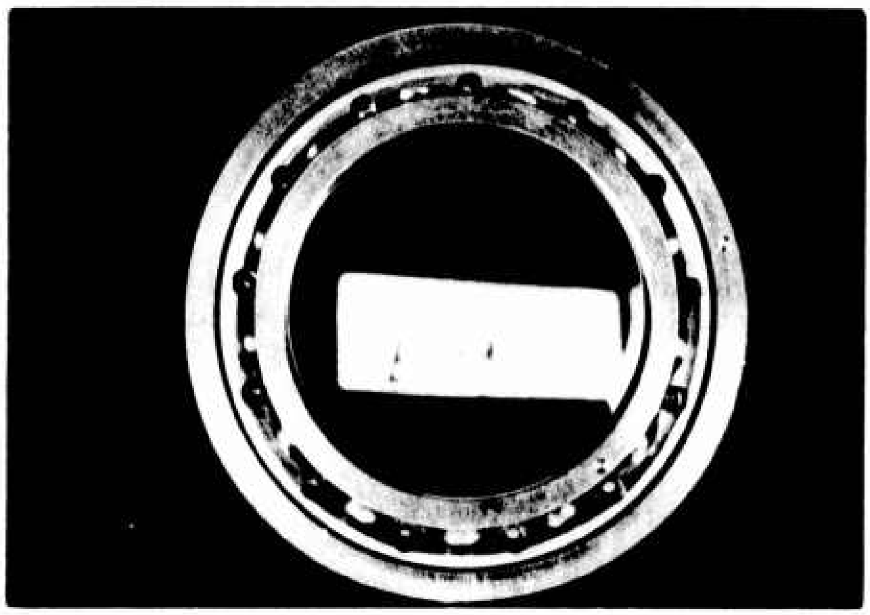


Figure 58. Bearing E19 Before Test (Outer-Land-Riding S-Monel Cage).

BREAK IN CAGE

Figure 59. Bearing E22 After Cage Failure, Test 8.

Figure 60. Bearing E21 After Lubrication Failure, Test 6.

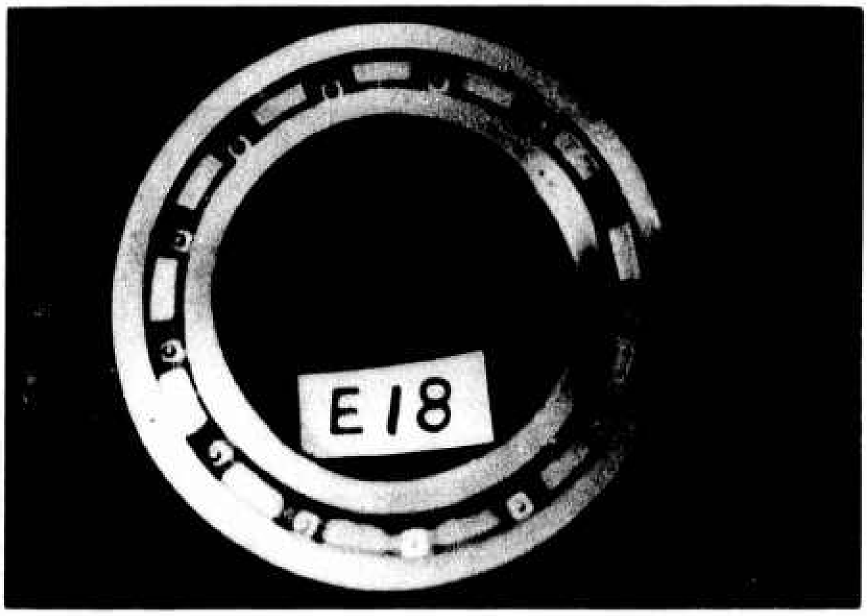


Figure 61. Bearing E18 Before Test (Modified Insert-Type Cage).

Figure 62. Bearing E18 After Failure, Test 10.

Figure 63. Bearing E24 After Failure, Test 9.

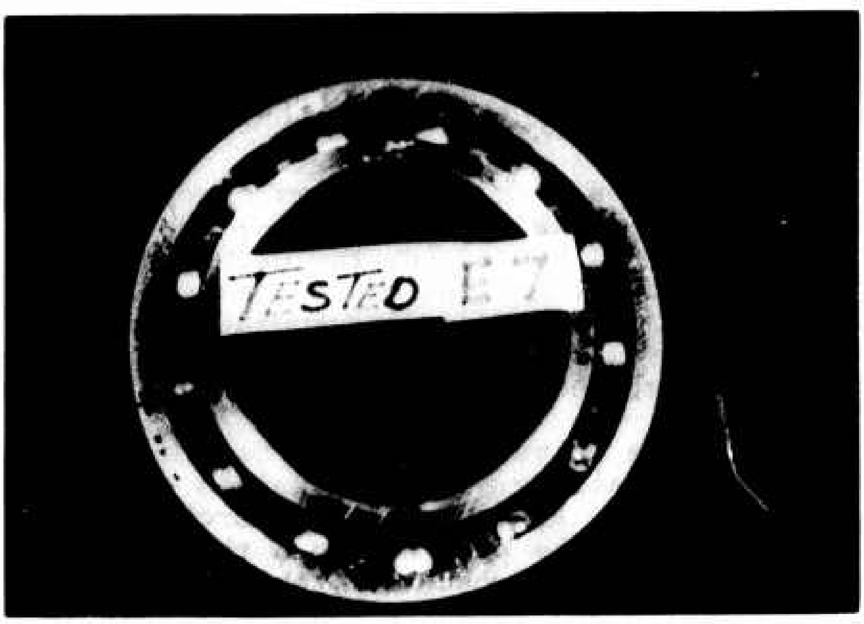


Figure 64. Standard Bearing After Test 11.

Both bearings ran 300 hours without regreasing. After 100 hours, the drive-end bearing ran somewhat hot (200°-230°F) most of the time, indicating significant cage rubbing.

The outboard bearing (S/N E20) with Design 3 cage (Figure 65 before test, Figure 66 after test) wore through the silver and into the base metal in the land contact area to depths of 0.007 inch on one half and 0.015 inch on the other. The O.D. of the cage had not contacted the outer lands. Balls had not worn through the silver on the pockets. There appeared to be a trace of Teflon left on the inner lands, embedded in the surface finish.

The inboard bearing (S/N E15) with Design 4 cage (Figure 67 before test, Figure 68 after test) wore to a maximum depth of 0.025 inch in the land riding surface, and this surface broke through. The O.D. of the cage had contacted the outer lands 180° from the area of heavy wear. Balls had not worn through silver in the pockets. The inner lands had no more than a trace of the Teflon coating left.

Rig configuration 4 had been quite successful in limiting grease purging, and the design remained relatively constant for the rest of the tests. The outboard end had been observed to purge during the first half hour; thereafter, neither housing purged significantly and both housings were still well supplied (4 to 5 ounces out of the original 7 ounces) at teardown. This is particularly significant because the drive-end bearing ran relatively warm through most of the test.

Test 13

This test evaluated a silver-plated machined steel cage design in bearings S/N E6 and E23 with Teflon-S coated inner lands. Both bearings ran 300 hours, but the drive-end bearing had a broken cage, and noise indications suggest that cage failure commenced at 140 hours. Cages incorporated annular grooves in the bore to assist grease circulation, but these plugged during test with a grease-wear debris mixture.

Figure 69 shows the drive-end bearing before test, and Figure 70 shows it after failure. The two halves of the cage had completely separated, and one half was broken into many pieces. The unbroken half of the cage had worn through the silver on both bore and O.D., with most wear, about 0.015 inch per side, on the bore. Balls had worn through the silver in the cage pockets. No more than a trace of Teflon remained on the inner lands. There were no grease additions to this bearing after start of the test, but the housing was nearly empty at shutdown.

Figure 71 shows the outboard bearing after test. This bearing

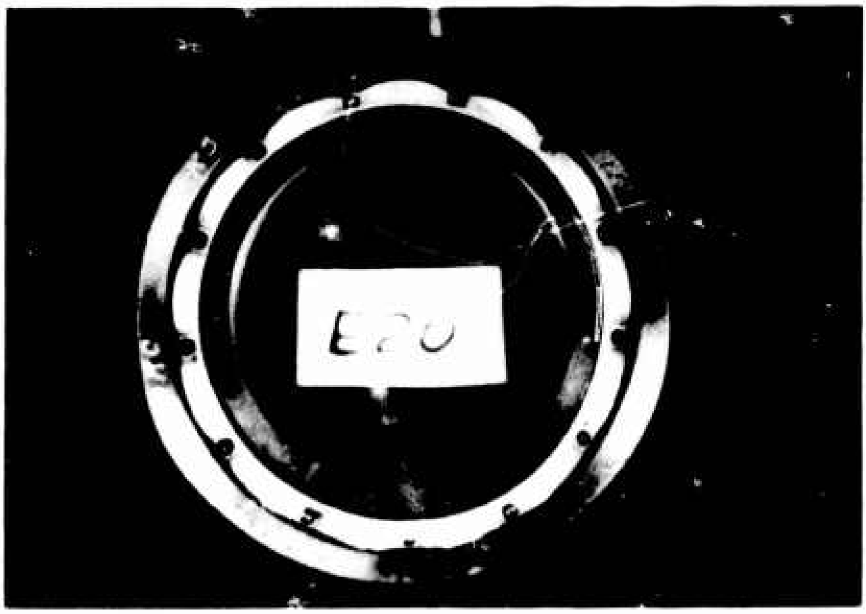


Figure 65. Bearing E20 Before Test (Cage Design 3).

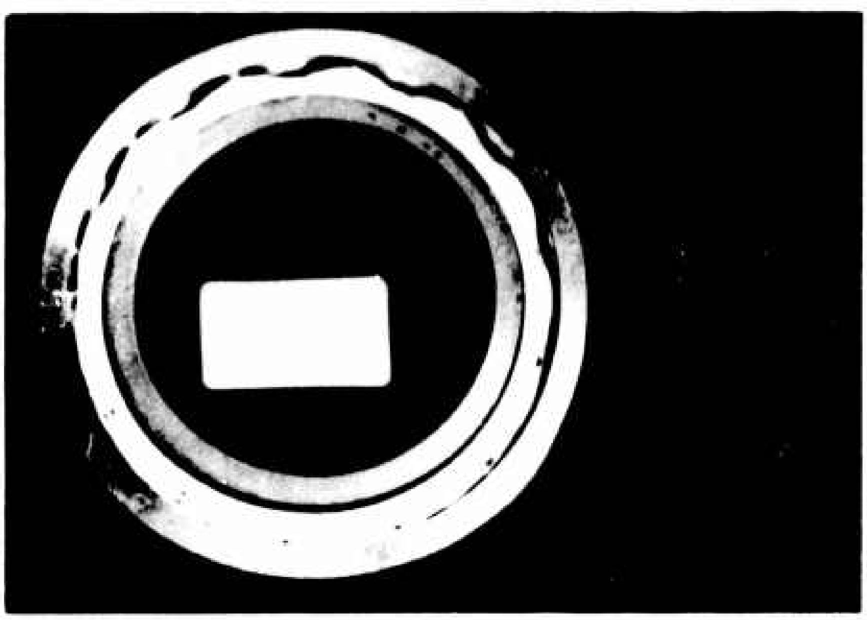


Figure 66. Bearing E20 After Test.

Figure 67. Bearing E15 Before Test (Cage Design 4).

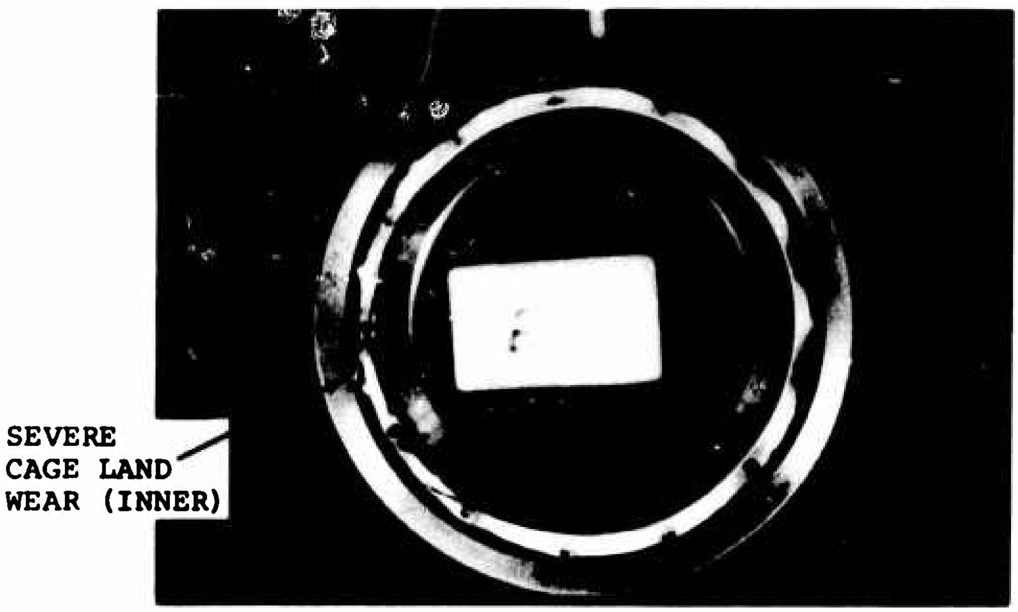


Figure 68. Bearing E15 After Test.

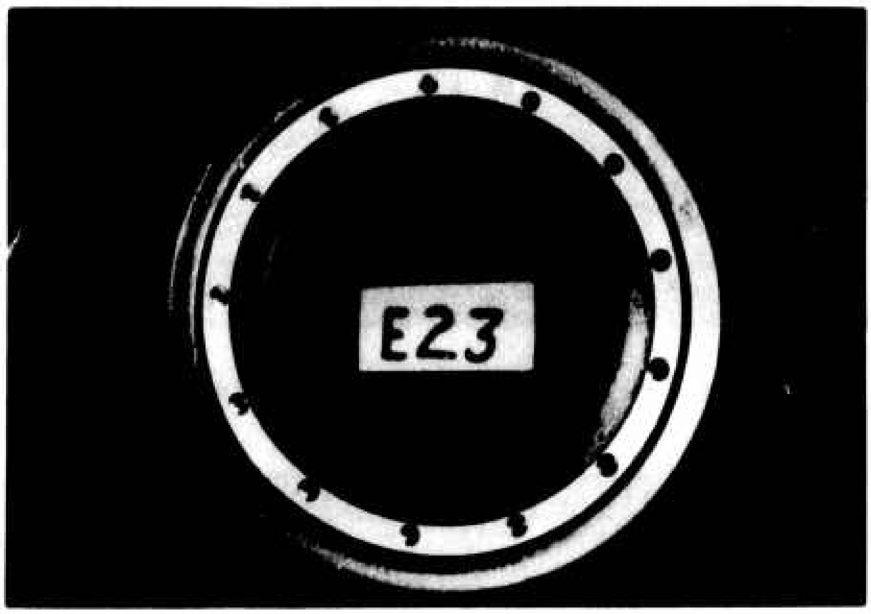


Figure 69. Bearing E23 Before Test (Silver-Plated Steel Cage).

Figure 70. Bearing E23 After Failure, Test 13.

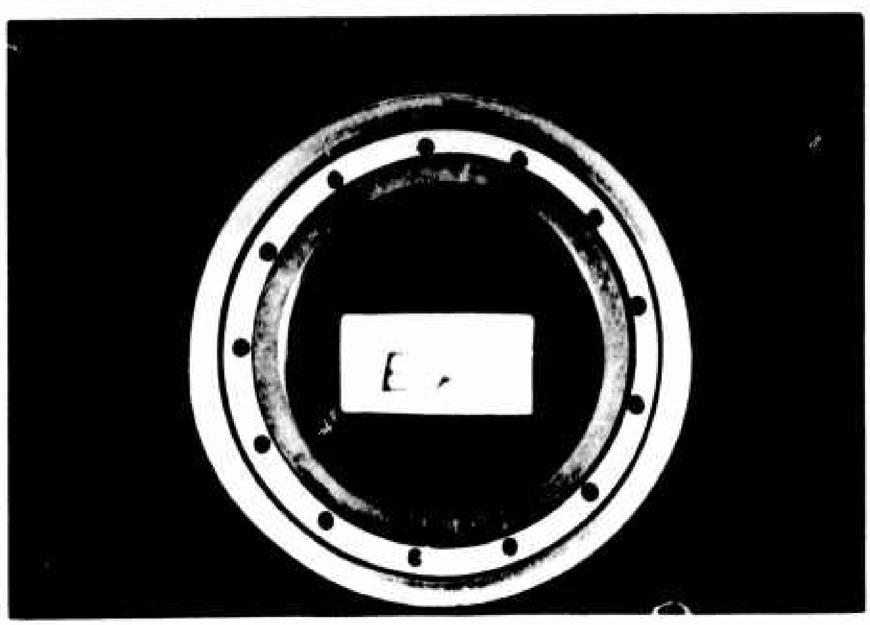


Figure 71. Bearing E6 After Test.

ran with considerable vibration throughout the test, but generally ran at 200°F or below. (Prior to each regreasing, however, it reached 295°-300°F.) It required regreasing with 3 ounces per time at 70, 135.8, and 225.1 hours. At shutdown the housing was nearly empty. The cage had worn in one area to a maximum depth of 0.021 inch. At a point 180° away from the area of greatest wear, the O.D. of the cage had rubbed the outer lands but had not penetrated the silver. Balls had worn through the silver plate in the pockets. No more than a trace of Teflon was left in the surface finish of the inner lands.

This cage design proved to be inadequate. It appears that the grooves in the bore of the cage soon became plugged with centrifuged grease residue and wear debris and were then unable to assist in grease circulation. Apparently the machining of the grooves and the deep counterbores for the short rivets permitted stress concentrations which weakened the structure.

Some regreasing was required in this test, as noted above. It appears that vibration, which tended to pump grease around the flingers, was responsible. This vibration was mostly due to a slightly off-square mounting of the bearing on the shaft. It was recognized at the start of the test but could not be corrected without removing the bearing and possibly damaging it in doing so.

Test 14

Bearing S/N 31, with Teflon-S coating on the inner ring lands, was installed on the drive end of the shaft. Bearing S/N 32 was installed on the outboard end.

At the start of test some instability was observed in the drive-end position. The housing tended to misalign in an oscillatory manner, apparently due to close clearance between shaft and seals. Several adjustments were required to minimize this condition, after which operation was very smooth. However, at each startup and shutdown, there was a brief period of apparent resonance in which the drive-end housing oscillated sharply.

There was very little purging of grease at the start of test. After 72 hours, approximately 1-1/2 ounces of grease had purged from the drive-end housing. Very little grease purged from the drive-end housing after that, until 243.9 hours, when 3 ounces of grease were inadvertently added to the housing, forcing out some of the original supply.

The outboard bearing purged almost no grease except at additions. These additions became necessary because of high temperatures associated with apparent cage rubbing. After 144 hours of operation, the temperature of the outboard bearing

began to rise slowly but steadily. At 160.5 hours this bearing reached 260°F and the rig shut down automatically. Three ounces of grease were added to the housing. Further additions of grease, 3 ounces each time, were made at 208.2, 244.7, and 268.4 hours in response to temperature excursions of the outboard bearing. The test was suspended at 302.6 hours.

Upon teardown, both housings were found to be well supplied with grease. Most of this grease was black.

The cage in the drive-end bearing (S/N 31) had separated at the mating faces. In most cases the rivets had broken. In four cases the rivet had worn the hole radially until it came out through the O.D. of one-half of the cage. Most of the rivet holes showed serious wear.

It was impossible to determine from the temperature record when the cage failure occurred. Wear on the bore of the cage was quite uniform all the way around and was likewise similar from one half to the other, generally 0.002 to 0.004 inch deep. Most of the ball pockets had lost silver through 360° of contact area. After the cage halves separated, at least one-half maintained ball spacing. Both halves show signs of incidental contact but no depth of wear in normally noncontact areas.

The cage in the outboard bearing (S/N 32) had worn severely on the bore in one area, with maximum depth about 0.045 inch. At a point 180° away from the area of maximum wear, the O.D. of the cage had contacted the outer ring and worn to a maximum depth of 0.025 inch. These worn areas were essentially the same on both halves of the cage. Ball pocket wear was heavy in the circumferential direction but generally light in the axial direction.

In the outboard bearing, the balls had a discolored band and both races indicated that a skid had occurred sometime during the run.

A little of the Teflon-S coating still remained on the lands of S/N 31's inner rings.

Ball paths in both bearings were quite well defined, indicating combined loading plus misalignment, as would be expected.

Photographs of the bearings, after test, are shown in Figures 72, 73, 74 and 75.

Results of this test indicated that problems still exist in bearings for the HLH application.

Increasing raceway curvatures appeared to improve vibration characteristics and reduce heat generation, as long as bearings

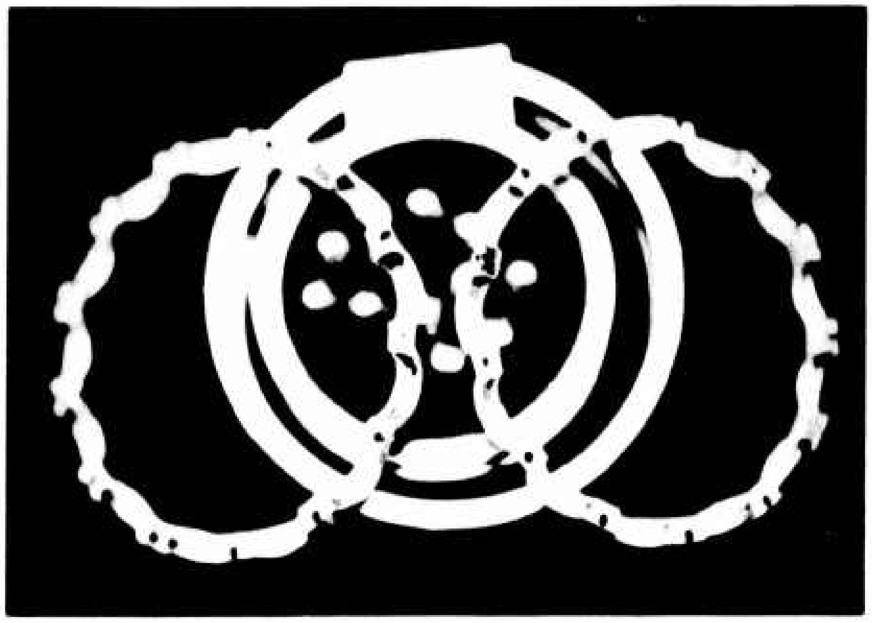


Figure 72. Components of Bearing S/N 31 After Test (Cage Was Broken).

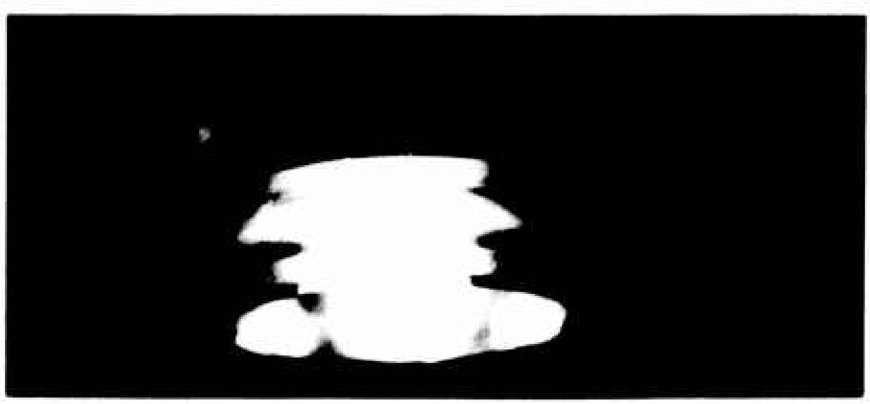


Figure 73. Inner Ring, Teflon-S Coated Lands of Bearing S/N 31 After Test.

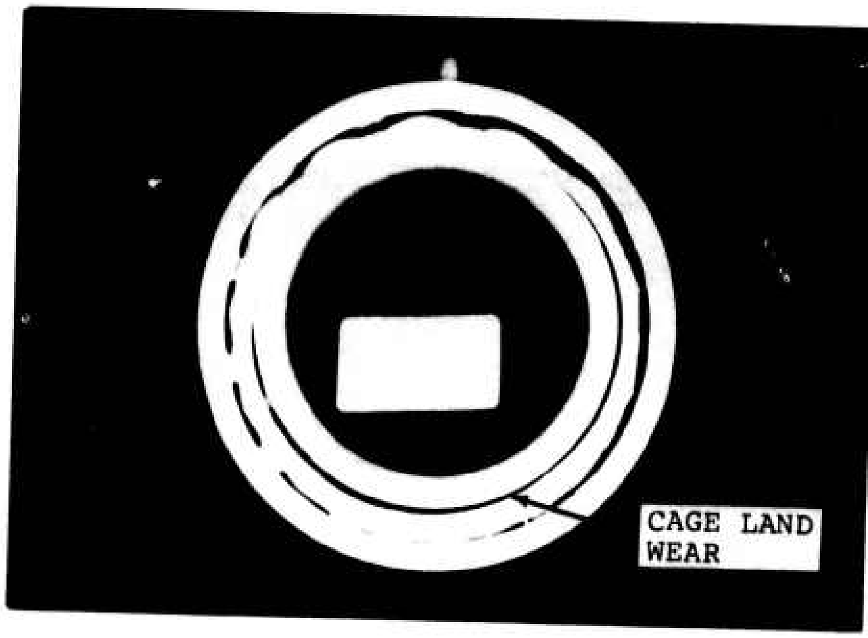


Figure 74. Bearing S/N 32 After Test.

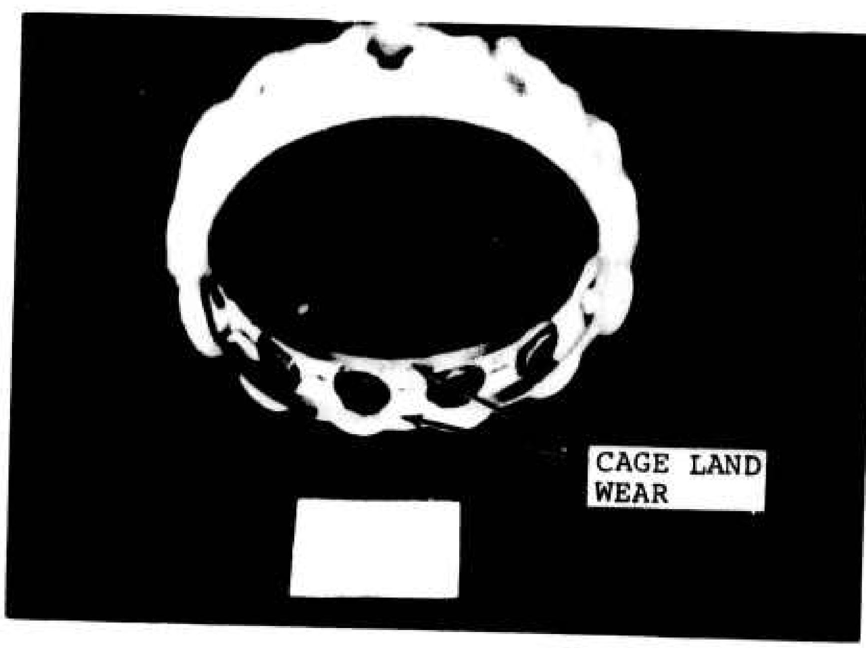


Figure 75. Cage S/N 32 After Test.

were functioning normally. Cage wear was essentially the same on both halves, again indicating an improvement.

Severe cage wear was caused by unbalance. For future testing cage balance should be checked and held to 3 gm cm maximum.

Ball forces in an axial direction were quite severe under misalignment, particularly during periods of instability. It appears logical to ease the situation by enlarging ball pockets in an axial direction.

The sealing method used in this test was quite successful. The need for additional grease in the outboard housing was not due to an overall shortage but rather to a highly localized situation caused by the unbalanced cage. This problem should be alleviated by balancing the cage.

Based on these test results, the following recommended changes were incorporated into additional test bearings:

1. The cage balance should be held to 3 gm cm maximum.

2. Heavier rivets are needed. It appears to be most practicable to reduce the ball complement by two and take advantage of the resulting extra space for heavier rivets.

3. The ball pockets should be enlarged in the axial direction.

4. The Teflon-S coating on the inner lands appears to be helpful and should be incorporated in the next design.

5. The present labyrinth seal shaft flinger arrangement should be maintained.

6. The 53% inner race curvature 54% outer race curvature should be maintained in future bearings.

<u>Test 15</u>

This test was performed to evaluate the Design 5 cage, plus variations in overall bearing design. Significant features changed were:

- 53% inner, 54% outer raceway curvatures

- Sermetel 72 coating on inner lands (replaces Teflon-S coating)

- 11 balls per bearing

- Heavier rivets in the cage
- Balanced cages
- Wide ball pockets

At startup, the outboard bearing (S/N 1) ran hotter than anticipated (up to 198°F) without external heat, then cooled to about 170°F, within 2 hours. The drive-end bearing (S/N 2) ran at less than 150°F. Heat was applied after 2 hours, and the rig ran quietly until shutdown at 47.1 hours. The outboard bearing temperature continued to drop in spite of some external heat (one infrared lamp per housing), to the 150°F range.

After restart at 47.1 hours, both bearings ran cool (130°-140°F) and additional heat lamps were applied, bringing both bearing temperatures to the 180°F range. The outboard bearing tended to run a few degrees warmer than the drive-end bearing through 120 hours, after which the drive-end bearing started to heat up.

By 130 hours, the drive-end bearing had increased to 200°F and continued to run between 200° and 240°F until 140.7 hours, when a power failure occurred. The power failure happened during unattended operation; when power was restored 35 minutes later, heat lamps came on but the motor did not. Housings and bearings soaked for 7-1/2 hours at temperatures recorded from 195°-255°F.

Bearing and housing temperatures dropped to normal levels when the rig was restarted, but the drive-end bearing soon began to overheat again:

Hours	Temp (°F)
142	200
146	290
146 - 156	270 - 300
156.2	220
156.4 - 158	290 - 300
158.4	240
158.9 - 159.1	290
160.5 - 167	200 - 220

It is probable that the cage broke during the excursion at 159 hours. At 167 hours, a check of the rig revealed that the labyrinth seals on the drive-end housing were loose and contacting the shaft. The rig was shut down, and it was necessary to disassemble the housing to make repairs. Temperature records indicate a sudden shift in housing temperature at 165 hours, suggesting that the seals came loose at that time.

The outboard bearing had run very consistently at 180°F until approximately 160 hours. Then its temperature increased sharply to 240°F, followed by a gradual increase to 265°F at shutdown.

The drive-end bearing's cage had parted at the rivets. Two broken rivets remained in each cage half; these had pointed ends, apparently due to rotation. Somehow, one or both of the cage halves had still accomplished ball separation until shutdown. The land riding area of the cage was worn all the way around, but preferentially to a depth of about 0.020 inch in one area. The O.D. of the two cage halves had also worn all the way around. In most areas the wear on the cage bore was not great enough to permit contact of the cage O.D. with the outer lands, so the O.D. wear must have occurred after partial cage fracture and separation.

Cage pockets were heavily worn, with a severe peening action having occurred near the split. The balls had made heavy contact in all pockets 90° from the direction of rotation, indicating that the cage had vibrated axially. This suggested that increasing the width of the pockets to provide greater ball freedom had not helped bearing performance.

Balls were straw color and were worn undersize about 0.003 inch.

Inner and outer races of the drive-end bearing looked surprisingly good, with a few small skid marks at the bottom of the inner race. Both race paths were wide, indicating operation in a variety of thermal and clearance regimes. The Sermetel 72 coating on the inner ring lands appeared to be largely eroded. Figure 76 shows the components of bearing S/N 2 after test.

The outboard bearing was intact with all cage rivets in place. These were removed to permit examination. The cage had worn preferentially in one area of the bore to a depth of no greater than 0.005 inch. The O.D. of the cage had not contacted the outer ring. Ball pocket wear occurred all the way around (except on the one extra-wide pocket), but was not severe.

Outboard bearing races were in relatively good condition with well-defined ball paths. These paths presented a cloudy appearance and indicate ball migration under the combination load-misalignment conditions. The inner ring showed the Sermetel 72 still covering most of the land area (Figure 77).

Outboard bearing balls were essentially to size but indicated some scuff marks, part of which may have occurred during bearing disassembly.

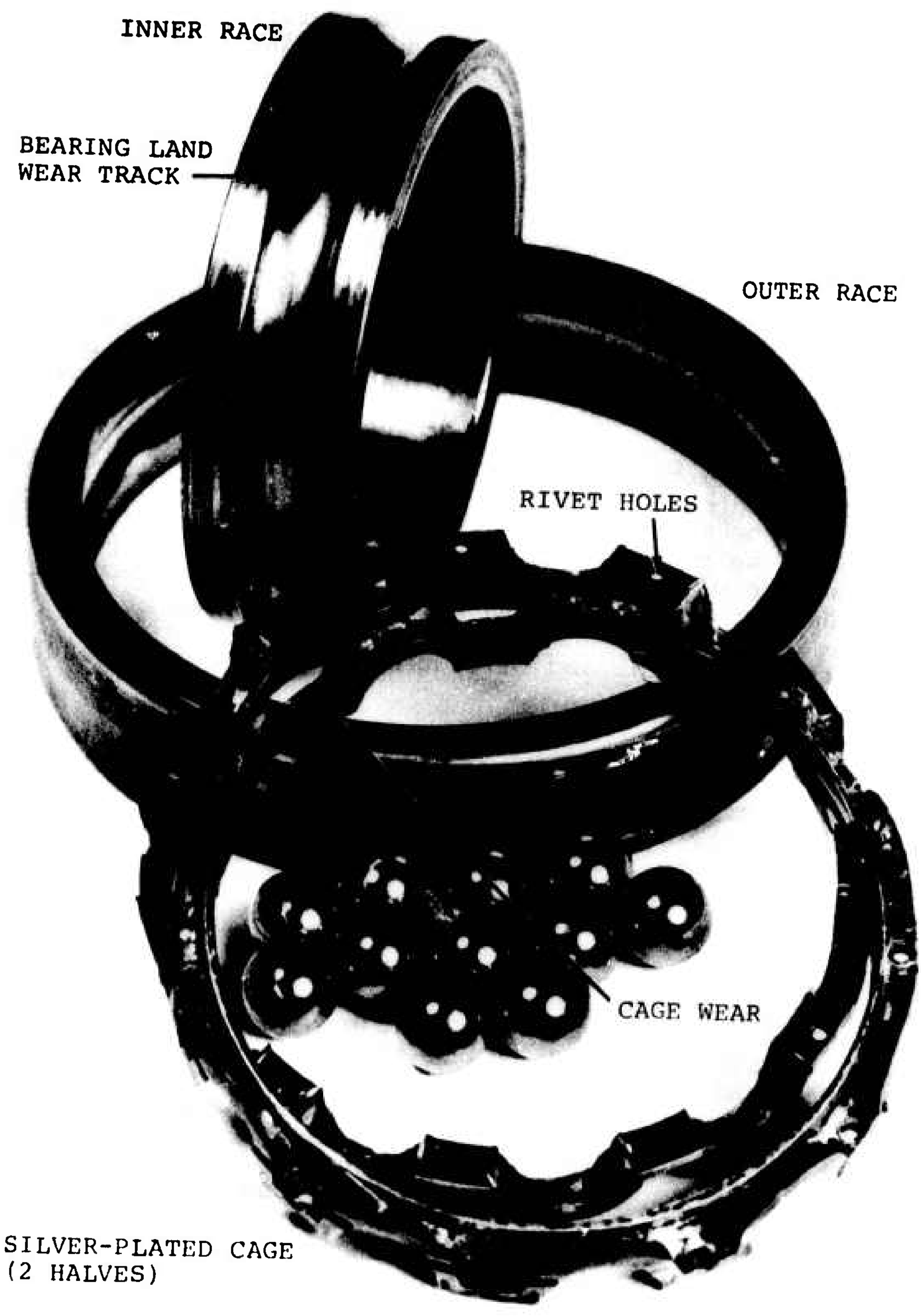


Figure 76. Drive-Shaft Support Bearing S/N 2, Drive End After Test.

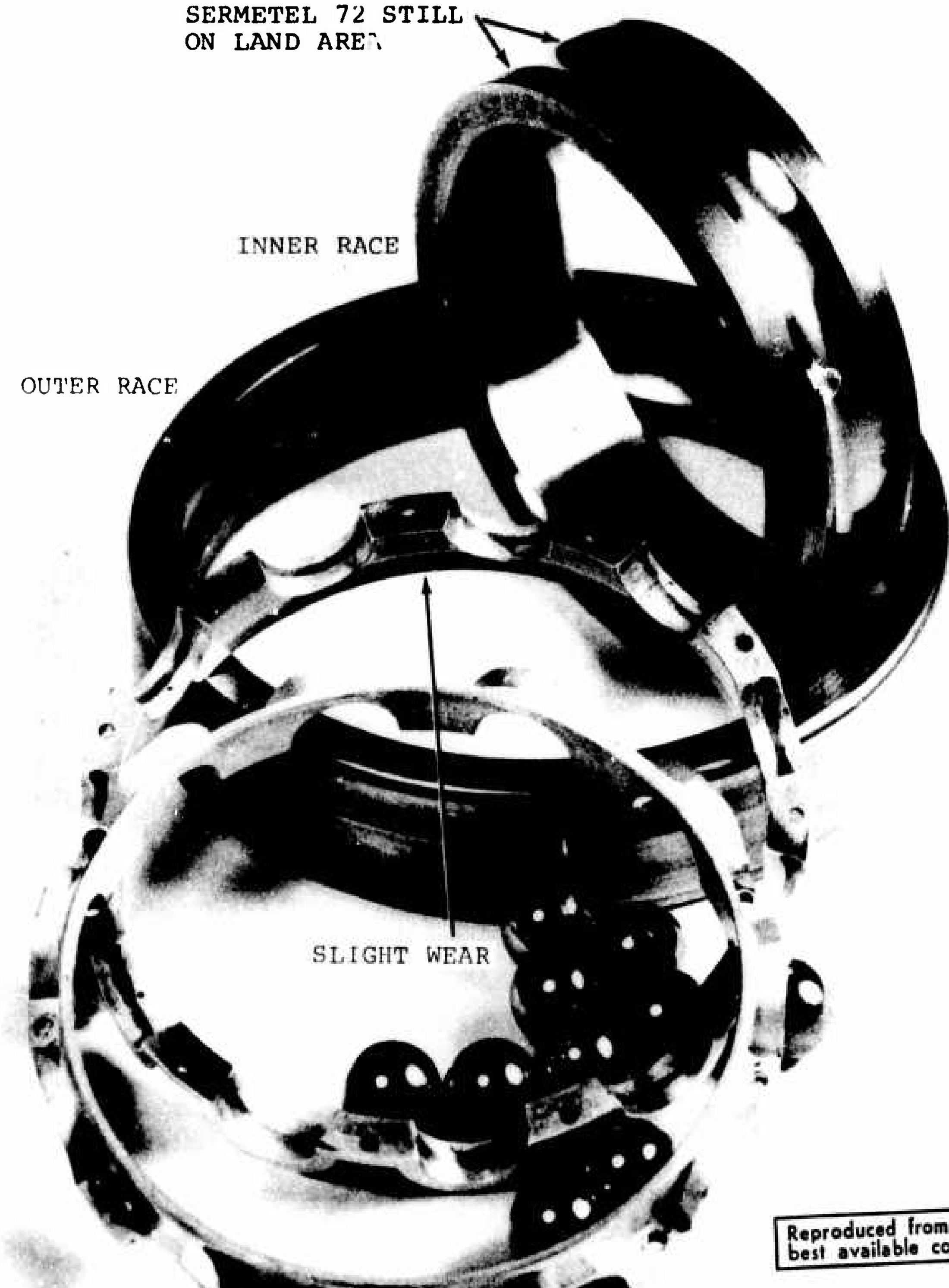


Figure 77. Drive-Shaft Support Bearing S/N 1, Outboard End After Test.

Both bearing housings were packed with 7 ounces of Aeroshell 22 grease before test, and no grease additions were made afterwards. Little grease purging occurred from either housing during most of the test, and no more than 1 ounce was lost by the outboard housing during the test. With the loss of the labyrinth seals on the drive end, most of the grease in this housing was lost and the drive-end bearing was rather dry. It is not possible to say how much of the drive-end grease was lost during temperature excursions and how much was lost at seal failure. The outboard bearing was well coated with grease, but its final temperature suggests that the bearing was not getting adequate lubrication at the time.

All grease, inside or outside of the housings, was discolored from gray to black. Except for the caked grease clinging to the drive-end bearing, all grease was moist, without indication of oil separation.

Test 16

After review of performance of the various cage designs tested, it was concluded that ball-riding cages were most promising if uniform control could be obtained. In order to overcome potential problems of inadequate strength in sheet-metal ball riding cages, a machined cage was designed as shown in Figure 24 and assembled into the bearing configuration shown in Figure 25.

Two bearings, S/N 3 and 4, were assembled with this cage. S/N 3 was installed in the drive-end position, and S/N 4 was placed in the outboard housing. Figure 78 shows bearing S/N 3 prior to test.

Prior to starting this test, the rig was modified to eliminate belt tension and slap as a test variable and to soften the mounting.

At the start of test, before external heat was applied, both bearings heated to approximately 150°F within 15 minutes, then cooled to 100°-110°F within 40 minutes. Heat was applied after 1 hour of operation. Between 4 and 6 hours, both bearings experienced temperature excursions to the 200°F range, after which temperature changes were very gradual and seemed to be related to ambient temperature. During the greater part of the 300-hour test, bearing S/N 3 ran at 190°-200°F, while S/N 4 ran at 180°F.

At the conclusion of test, the drive-end bearing was found to have purged nearly all of its grease, but internal surfaces of the housing, and the bearing, were still well coated. The outboard bearing had purged very little grease. Purged grease and residual grease were still in good condition; there was some discoloration in part of the purged grease from the drive

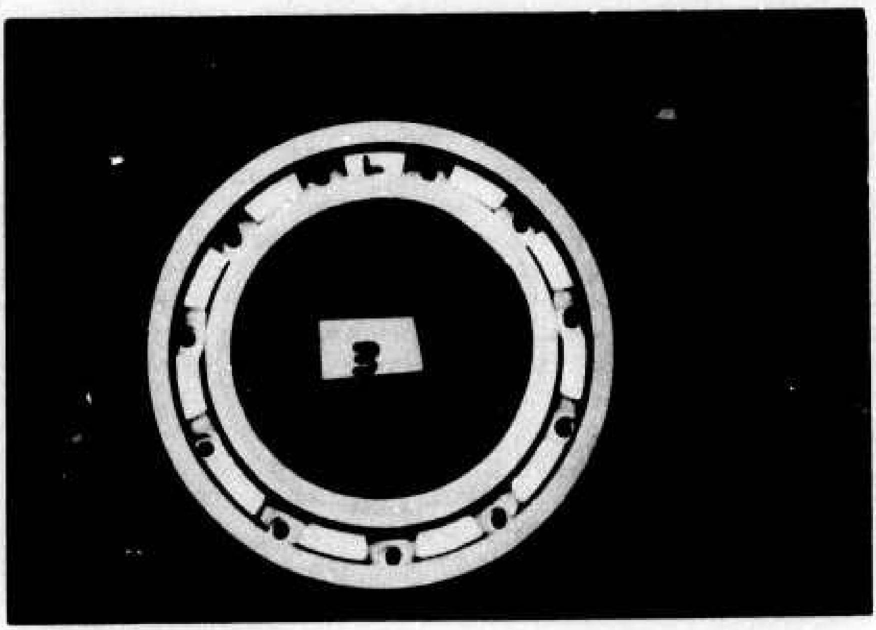


Figure 78. Assembled 111-KS Bearing S/N 3 Prior to Test.

end, probably from passing through the seal. No grease was added to the bearing housings after start of test.

Both bearings were in serviceable condition at the completion of the test. Balls from both bearings indicated banding, with areas of "original" finish and dull finish. In bearing S/N 3, all balls looked much the same. In bearing S/N 4, there was variation from ball to ball. Ball paths were clearly distinguished but showed no depth of wear.

The only wear in both cages occurred at the ball contact near the bores. In neither bearing had the balls worn through the silver plate. In bearing S/N 3, the ball pockets were polished equally on both sides. In bearing S/N 4, the wear was heaviest on one side; at the very bottom of some of the pockets on the worn side, silver had broken loose at the corner.

Bearing S/N 3 showed brinelling after disassembly, and balls showed scratches. These effects were superimposed upon running tracks and obviously occurred after running was complete. The cage was removed by grinding off rivet heads, and brinelling probably occurred during this operation. Another possible cause was removal of the bearing from the shaft. Bearing S/N 4 was removed from the shaft in the same manner, but it was disassembled by drilling rivets; it did not show significant brinelling.

Prior to test, both bearings had radial clearances of 0.0017 inch. After disassembly, measured clearances were 0.0013 and 0.00165 inch on bearings S/N 3 and 4, respectively.

Tests 17 and 18

As the result of the successful operation of bearings S/N 3 and 4 in the previous test, it was decided to run four more bearings of the same design to obtain reliability data.

The rings in bearings S/N 1 and 4 had been run previously in this program, S/N 1 with a Design 5 cage. Both bearings were rehoned and outfitted with new balls and cages. All four bearings had Design 6 cages. Bearings were installed in the two test rigs as follows:

	Machine A	Machine B
Drive End	S/N 5	S/N 6
Outboard End	S/N 4	S/N 1

Both rigs ran continuously for 300 hours with no regreasing. No bearing purged significantly, and all housings were well supplied with grease at the end of test. After the scheduled shutdown, machine A was noisy during coastdown.

Bearings S/N 1, 5 (Figure 79), and 6 were found to be in excellent condition. Radial clearances were 0.0019, 0.0017, and 0.0019 inch, respectively, compared with 0.0019 inch for all bearings before test. Cages were effectively supported by balls, so no contact had been made with ring lands. These bearings were not disassembled.

On rig disassembly, grease adjacent to bearing S/N 4 was observed to be black but soft and oily, indicating contamination with wear particles. The cage was no longer ball riding, and contact had been made with inner and outer ring lands. Radial clearance had increased from the original 0.0019 inch to an erratic 0.0035-0.0043 inch. This bearing was disassembled and the following conditions were observed:

1. Ball pockets were still contoured, but ball drop, per pocket, had increased by 0.020-0.025 inch, permitting contact between cage and lands (Figure 79).

2. Balls were out-of-round, varying in diameter, within a single ball, from 0.4045 to 0.4062 inch.

3. Balls showed visible bands.

4. Cages had worn through the silver-plate on both bore (Figure 80) and O.D. in some areas. Actually, wear on the cage rails was relatively uniform, ranging from practically nothing to about 0.003 inch.

5. While all rivets had been intact and apparently solid, a small degree of fretting could be seen at the interfaces of the two halves.

6. Ball pockets were heavily worn, with silver removal in substantial areas. Pocket wear was concentrated fore and aft, with only light contact axially.

7. The inner raceway showed two distinct ball paths, one indicating essentially pure radial load and the other thrust. Both paths indicated some minor surface distress, with a cloudy appearance.

8. The outer raceway also showed two ball paths, although not as distinctly as did the inner race. In addition, the outer race showed scallops; these markings were not deep enough to be perceptible to touch.

9. There had been no gross overheating.

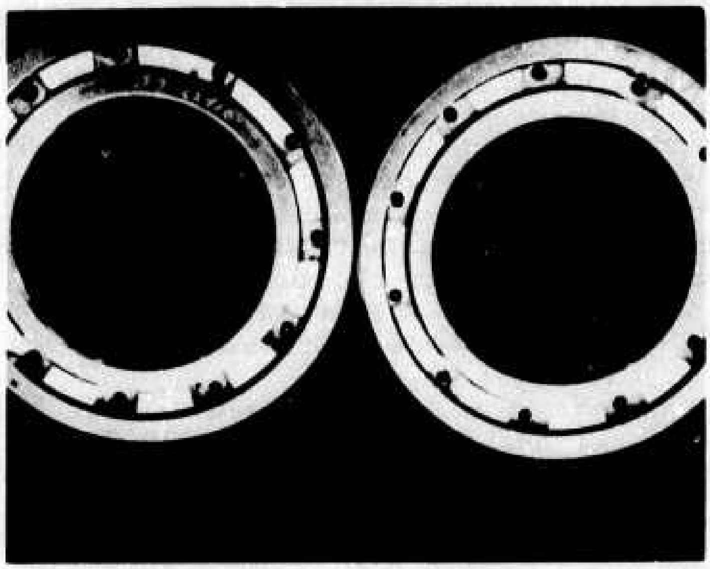


Figure 79. Bearings S/N 4 and 5 After Test (S/N 4 on Left Showed Some Cage Wear).

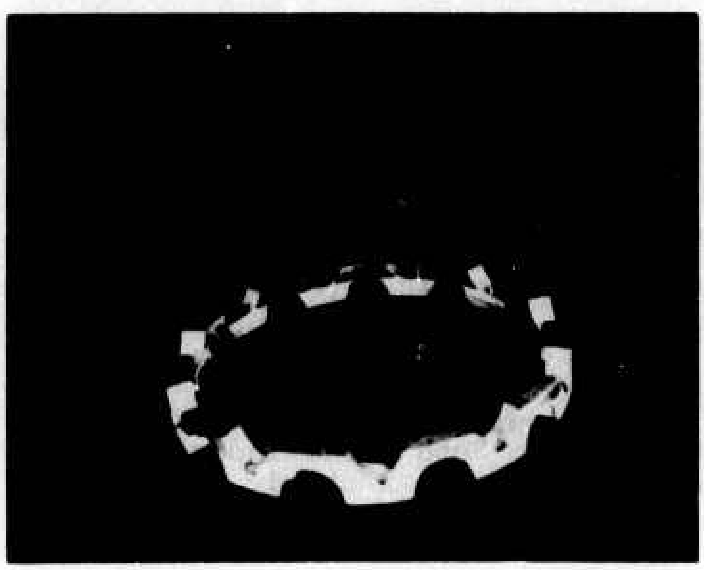


Figure 80. One-Half of Cage From Bearing S/N 4 After Test, Showing Wear.

The damage to bearing S/N 4 did not constitute a bearing failure. It appears that wear became regenerative with particles being trapped in adjacent grease and tending to further abrade the bearing. While the housing had a great surplus of grease, the grease tended to stratify. Periodic regreasing, which would cause old grease to purge and flush the bearing, would probably have been helpful.

Cause of the bearing distress was apparently the very light load on the bearing which permitted balls to skid at times. Skidding under lightly loaded conditions is a common, but not completely predictable, phenomenon. Bearing S/N 4, with identical design, ran in the same housing and same position on the same rig in a previous test without experiencing difficulty. In machine B, bearing S/N 1 also ran under the same applied conditions without damage.

Temperatures for tests 17 and 18 are summarized as follows:

Bearing S/N 1

Temperature increased to 150°F within 15 minutes after start, then cooled to about 110°F. After heat was applied, temperature increased to 200°F, then cooled to approximately 170°F, after about 60 hours running, there were three brief excursions to 200°-210°F. Other brief excursions occurred at 74 hours, 79 hours, and 146 hours, with the latter reaching 238°F. During the remainder of the test, and between excursions, the bearing ran from 165° to 180°F.

Bearing S/N 6

Temperature increased to 173°F within 15 minutes after start, then cooled to less than 100°F. After heat was applied, there was an excursion to 200°F, at 21 hours, followed by a return to normal operation. There were no further temperature excursions, and the bearing ran generally between 150° and 170°F.

Bearing S/N 4

Temperature increased to 172° at 40 minutes after start, then cooled to approximately 90°F. There was no temperature excursion when heat was first applied, but there were several brief excursions to the 210°-220°F range at 230 hours, followed by a return to normal. The bearing generally operated from 170°-190°F.

Bearing S/N 5

This bearing ran at 110°F after an unheated start with very slight excursion. Temperature excursions occurred after apparently stabilized operation, including:

Hours	Temp (^oF)
56	215
75	201
175	260

Temperatures exceeded 200^{o}F from 171 to 178 hours. Most of the time the bearing operated between 160^o and 180^{o}F.

SUMMARY OF TESTING

Of the several cage designs initially tested, only the machined inner-land-riding S-Monel cages consistently ran the scheduled 300 hours. All specimens of the five inner-land-riding S-Monel cage designs suffered objectionable wear, and bearings which operated on the drive end of the test rig consistently experienced greater wear than those on the outboard location. The Design 3 cage, incorporating silver-plate, experienced the least wear or damage of the cages tested prior to test 16.

Performance of pressed-steel cages, both with and without Teflon-S coating, was somewhat erratic. Two bearings ran until the 300-hour scheduled suspension; one cage failed at 110 hours; another cage failed at 246 hours. One bearing which ran the scheduled 300 hours showed overheated ball pockets. While the pressed-steel cage has the advantage of minimum bulk and permits a maximum grease fill, it does not have the reliability which the application demands.

The control of manufacturing tolerances of this type of cage design appears to be the major disadvantage.

The fail-safe cage with DuPont Vespel SP-21 inserts appeared to be too bulky and too difficult to lubricate for this application. A basic problem was the small section of the MRC 111-KS ball bearing. Machining operations on SP-21 material require some minimum section for structural stability. When space within the bearing package is limited, the supporting frame or the space for lubricant is seriously reduced. The original design did not permit adequate lubrication. When this design was modified by removing sufficient stock from the frame to significantly improve lubricability, the structural strength was impaired and cage failure occurred. In a heavier section bearing this fail-safe concept could be quite satisfactory.

The steel cage and the outer-land-riding S-Monel cage designs did not have sufficient structural strength for the application and failed during its testing.

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56	215
75	201
175	260

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The steel cage and the outer-land-riding S-Monel cage designs did not have sufficient structural strength for the application and failed during its testing.

At the completion of test 15, a review of all the test data was conducted to determine what changes could be made to eliminate the serious cage wear problem. A review of each of the bearing temperatures, both unheated and heated, was conducted as shown in Table V. This review showed that an increase in race curvature resulted in lower operating temperatures and that the ball-riding, pressed-steel cage also provides good operating characteristics. Therefore, the next and final bearing configuration tested (Figure 25) included increased race curvatures and a machined ball-riding S-Monel silver-plated cage. Very little cage wear occurred during 300 hours of testing this design; therefore this design satisfied the HLH/ATC design criteria.

The final housing configuration (Figure 31) that provided adequate grease retention was a combination of labyrinth seals plus shaft flingers. Initial testing showed that grease circulating devices used in test rig configurations 1 and 2 (Figures 28 and 29) tended to work the grease while not necessarily improving bearing lubrication; therefore, these devices were eliminated early in this program.

CONCLUSIONS

This test program established that a Conrad type ball bearing operation at 632,000 DN with grease lubrication is entirely within the state of the art. However, housing design, bearing geometry, and cage design are very critical and greatly influence the performance of the bearing.

The following design criteria were established for the successful operation of the HLH/ATC engine shaft support bearing for 300 hours without regreasing:

- Aeroshell 22 grease conforming to military specification MIL-G-81322 is a satisfactory lubricant for the HLH application.

- Misalignment presented the greatest hazard to successful bearing operation in this program; consequently, the bearing raceway curvatures were modified (increased to 53% on the inner surface and 54% on the outer surface) to reduce the effects of misalignment on cage wear, raceway distress, and reduced friction losses.

- The use of shaft flingers and labyrinth seals provides adequate grease retention within the bearing housing cavity. However, excessive vibration or temperature may cause some migration of grease past the flingers and seals.

- The use of standard regreasing procedures, such as pumping grease through a grease fitting until grease is purged through seals, is a satisfactory method of regreasing the bearing. Brief temperature excursions do occur after regreasing, but temperature stabilization is achieved after a short period of operation.

- The best cage configuration is a silver-plated, machined, ball-riding S-Monel cage. The cage should be balanced prior to assembly to insure satisfactory operation.

Based upon the above design criteria, the HLH/ATC engine shaft support bearing and housing shown in Figure 81 were designed. Boeing Vertol will test this design for several hundred hours prior to installation in the prototype HLH helicopter.

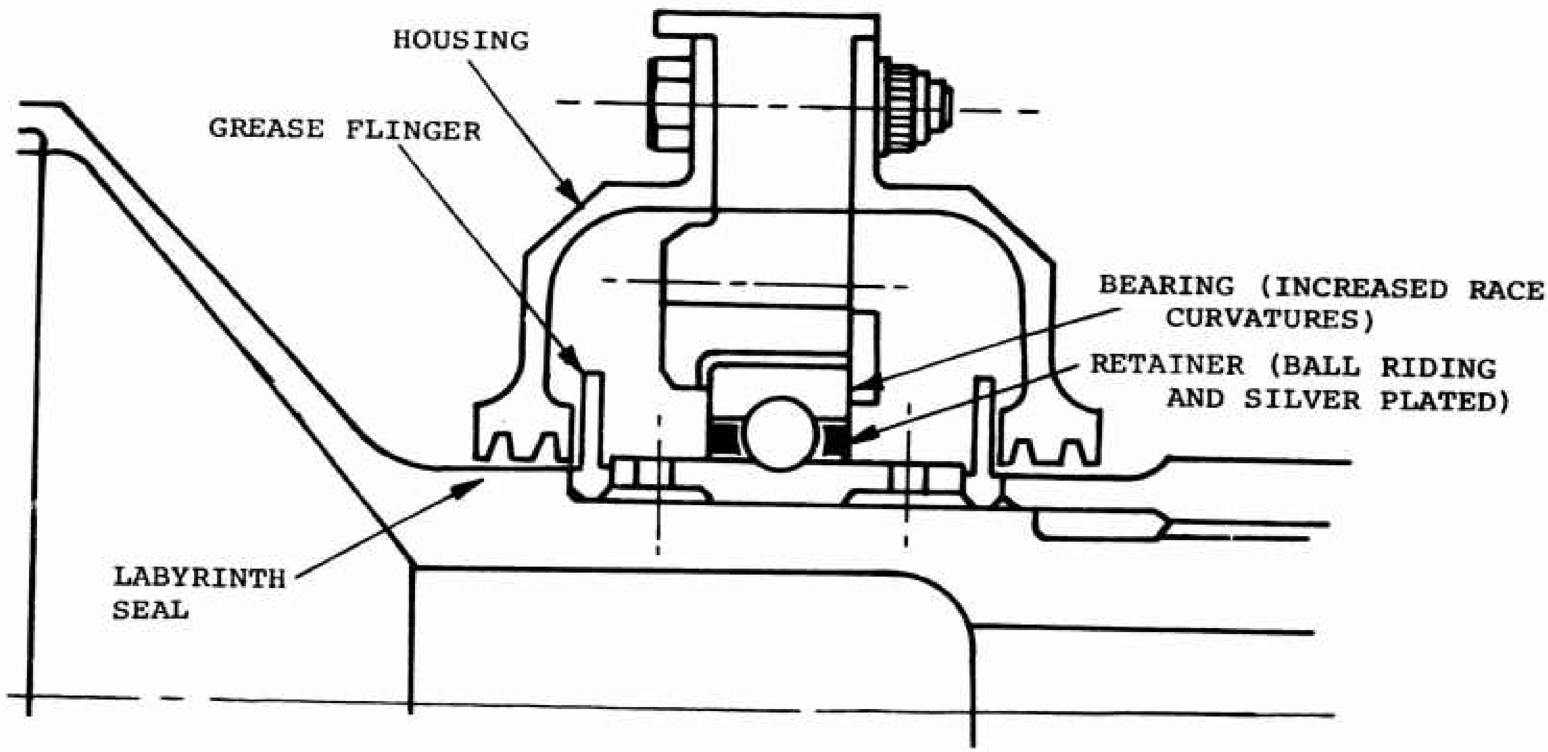

Figure 81. Final HLH/ATC Engine Shaft Support Bearing Configuration.

RECOMMENDATIONS

Full-scale test rig testing of the HLH/ATC engine shaft support bearing and housing has demonstrated that a ball bearing can operate at a speed of 632,000 DN for 300 hours without regreasing. It is recommended that additional testing be conducted for an extended duration to determine bearing performance and cage wear as well as the effect of regreasing. Also, the bearing should be operated in various loading and shaft angles to determine the effects of load variation on bearing performance. Additional work should also be directed toward providing a better shaft seal to minimize grease purging under severe vibration and temperature excursions.

APPENDIX: TEST RESULTS OF ENGINE SHAFT SUPPORT BEARING ENDURANCE TEST PROGRAM

BACKGROUND

Test bearings conforming to Figure 25 (MRC-111KS) were residual from previous testing. Three bearings were put into test with no processing, except for cleaning, after having operated 300 hours under the previous test program. The fourth bearing was assembled from a cage and rings which had also run previously.

The test rigs were the same as used under previous testing, incorporating modifications to the drives and mountings which were established during testing. Machine B, however, was lowered from its inclined orientation to be horizontal. Schematic of the test rig is shown in Figure 31.

Two bearings, S/N 1 and 6, were run in Machine A with test duration scheduled to be 1000 hours. Regreasing was scheduled at 300 hours and 600 hours with cursory bearing inspection at those times. Overheating of bearing number 7, in the drive end position, caused modification of the schedule with more frequent examination and regreasing of one bearing. At 871.0 hours, bearing number 6 was replaced in test by bearing number 5.

Two bearings were run in horizontal Machine B for 300 hours without regreasing. One of the bearings was serial number 5. The second bearing was assembled from the following:

Inner Ring	- S/N 32	Modified and tested under previous Test Program.
Outer Ring	- S/N 31	
Cage	- "No. 0", previously run 300 hours in S/N 4	
Balls	- New	

The races of #31-32 were rehoned to remove indications of previous running. Radial clearance was .0017 inch.

TEST EQUIPMENT AND PROCEDURE

The basic test equipment used is shown in Figure 31. Two test rig spindles were modified slightly by increasing diameters of shaft flingers and increasing clearance at the labyrinth seals. Also, the drives were modified to remove belt

tension as a possible variable; and installation of elastomeric mounts was changed to simulate more closely the arrangement used in the application.

Both spindles were used in this program. Machine A was set up in the configuration described above, and run under the following conditions:

Shaft Speed	11,500 rpm
Shaft Misalignment	0° 15'
Shaft Inclination	34°
Shaft Weight	80 lb
Shaft Unbalance	0.10 ±.02 inch-ounce
Lubrication	Aeroshell 22 grease
Ambient Temperature	180°F

The spindle was mounted so that upper bearing received substantially all of the axial load.

Duration	1,000 hours, with regreasing

Machine B incorporated the same modification as Machine A, but its base was shifted so that the spindle was horizontal. Its operating conditions were:

Shaft Speed	11,500 rpm
Shaft Misalignment	0° 15'
Shaft Weight	80 lb
Shaft Unbalance	0.10 ±.02 inch-ounce
Lubrication	Aeroshell 22 grease
Ambient Temperature	180°F
Thrust Load	essentially none on either bearing
Duration	300 hours without regreasing

Each test started without external heat to determine self generated temperature. After stabilization, two 250-watt infra-red lamps were turned on each housing and the rig was covered with an insulated box. These heat lamps plus bearing heat and insulation were sufficient to achieve 180°F ambient temperature. It was sometimes necessary to turn off one or more lamps, depending upon bearing heat development.

1000-HOUR TEST

Test bearing S/N 1 and 6, conforming to Figure 25, were installed in Machine A. These bearings had both been run 300 hours under the same test conditions in Machine B.

Both bearings had been washed and examined visually after completing the 300-hour test, but had not been disassembled.

Both bearings had .0019 inch radial clearance after that test, the same as before. Bearing S/N 1 was installed in the outboard end, with bearing S/N 6 in the drive-end of the spindle. These were the same orientations as in previous testing, and direction of rotation was the same.

Bearings were scheduled to run until failure, or 1000 hours, with inspection and regreasing at 300 and 600 hours. Inspection was limited to examination of a bearing without removal of the bearing from the shaft and without cleaning; in each inspection, a labyrinth seal was removed to permit observation of the bearing. Regreasing was accomplished by a grease gun through a fitting on the housing. Seven and one-half ounces of grease was added to a housing at each regreasing, tending to purge residual grease from the housing and to flush used grease from the bearing.

Both bearings operated satisfactorily through the scheduled shutdown at 300 hours. After 240 hours running, the drive-end bearing, S/N 6, ran consistently in the 190°F range. After 290 hours running, the outboard bearing, S/N 1, ran hotter than 180°F with two brief excursions, one to 225°F at 295 hours and one to 255°F at 299 hours.

At the 300-hour inspection, it appeared that no more than 2 ounces of grease had been lost from the outboard housing, and the remaining grease was in excellent condition. The temperature excursions which this bearing experienced shortly before shutdown were apparently due to momentary depletions adjacent to the bearing with subsequent replenishing from the housing. No indication of wear debris was observed in the bearing, and the cage appeared to be in excellent condition.

Only 1-2 ounces of grease remained in the drive-end housing. The residual grease was slightly discolored but retained good lubricating properties. The bearing appeared to be in good condition, with the cage still ball-riding, although the grease discoloration indicated some internal wear.

After regreasing at 300 hours, the rig was restarted. At 497.2 hours, the drive end bearing heated to 265°F, causing automatic shutdown of the machine. During this run, both bearings generally operated in the 180°-190°F range; after 485 hours, the drive end bearing experienced a number of excursions to the 220°F range, before finally heating to 265°F.

The drive-end bearing was examined and regreased following the shutdown at 497.2 hours. Only 1-2 ounces of grease remained in the housing, and this grease was visibly darker than the residual grease in the same housing at the 300-hour inspection. The bearing cage was intact and still ball-riding. However ball pockets had worn sufficiently that in some

orientations the cage bore almost contacted (approximately .002 inch clearance) the inner lands. The outboard bearing was not examined or regreased at this time.

After restart at 497.2 hours, the drive-end bearing heated to 220°F and ran at 200°-220°F for about 8 hours before cooling to the 180°F range. At 637.8 hours, the machine was shut down to permit inspection of both bearings. This inspection and regreasing replaced that which was scheduled for 600 hours. At shutdown, both bearings were performing normally with no indication of distress.

The outboard housing had lost 4-5 ounces of grease; the drive-end housing had lost about 6 ounces. The residual grease near the outboard bearing was slightly discolored, and that near the drive-end bearing was dark, similar to that observed at the 497.2-hour examination. The outboard bearing was apparently in excellent condition. The cage of the drive-end bearing did not appear to be making contact with the ring lands, but clearance was less than .002 inch.

Both housings were regreased after the 637.8-hour inspection. After restart, without applied heat, the outboard bearing ran at 108°F while the drive-end bearing ran at 162°F. After heat was applied, both bearings ran normally for about 100 hours. Then the drive-end bearing began a temperature excursion which terminated in an automatic shutdown when the bearing reached 260°F at 745.9 hours. The drive-end bearing was found to have lost about 6 ounces of grease, and that remaining in the housing was dark. The bearing cage was contacting the lands in some orientations; when the housing was rotated, the cage was observed to shift radially, suggesting a pumping action.

Both labyrinth seals on the drive-end housing and the outboard seal on the outboard housing were examined at this time. All were found to be severely worn, with total clearance of .040 - .045 inch, compared with an initial clearance of .020 inch maximum. The three seals were replaced by in-tolerance seals of the same design - two used and one new.

It was necessary to remove the drive-end bearing from the shaft for seal replacement.

The drive-end housing was regreased. The outboard housing was disturbed somewhat during replacement of the labyrinth seal but no additional grease was put into it.

After restart of test, the drive-end bearing heated initially to 212°F, then ran for 35 hours at 180°F without heat lamps. (Heat was applied to the outboard housing, however, and this had some effect on the drive-end housing). Thereafter, one heat lamp was required on the drive-end housing to maintain temperature.

At approximately 852 hours, the drive mechanism on the temperature recorder failed. The instrument stopped recording and made the automatic shutoff device inoperative. At that time bearing operation was normal. When the test rig was checked, at 871.0 hours, the drive-end bearing was found to be running at 339°F. The machine was shut down and bearings were inspected.

The drive-end bearing was found to be coated with baked grease and to be stiff. Less than 1 ounce of grease remained in the housing, and it was black, indicating presence of wear debris. The housing seals had become worn, to produce a clearance of approximately .040 inch. It was decided to replace the drive-end bearing, S/N 6, with the drive-end bearing from Machine B (bearing S/N 5) to complete the test. The one housing was then regreased and the test restarted.

Bearing operation during the final 129 hours of test was normal, with no temperature excursions. Before heat lamps were applied, the outboard bearing ran at 108°F and the drive-end bearing reached 152°F. The test machine was shut down at 1000 hours. Upon teardown, the outboard housing still held about 5 ounces of grease; this grease was dark brown. The drive end housing held 5-6 ounces of slightly discolored grease. Bearings were disassembled for examination.

EXAMINATION OF BEARINGS AFTER TEST

Outboard Bearing S/N 1

The radial clearance of this bearing increased from .0019 to .0020 inch during the 1000 hours of operation.

The cage was completely supported by the balls with no contact between bore or OD with inner or outer lands. There was some wear in the ball pockets, with silver removed in narrow bands near the bore of the cage, fore and aft. Location of polished areas in the ball pockets varied from one to another, indicating eccentric motion of the cage.

The races were in good condition with ball paths showing thrust load, radial load, and misalignment. Apparently the bearing received some thrust, although the mounting method tended to place all of this component on the drive-end bearing.

Balls were cloudy with some variation in intensity.

Figures 82 and 83 show components from bearing S/N 1 under magnification (3.2X and 2.4X respectively). The ball path in Figure 82 is mostly on one side of the inner raceway.

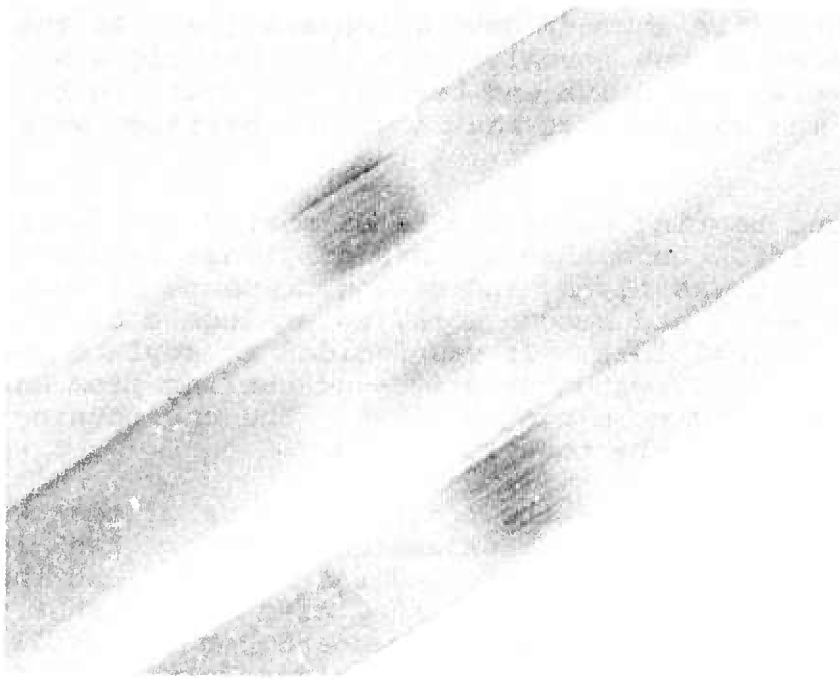

Figure 82. View of Inner Race, Bearing S/N 1.

Figure 83. View of Cage Pockets and a Ball, Bearing S/N 1.

Figure 83 shows wear in the ball pockets near the cage bore. Discolored areas in the pockets, toward the cage OD, are residual grease films.

Drive-End Bearing S/N 6

The radial clearance of this bearing increased from .0019 to .0022 inch during 871 hours of operation.

Grease had baked onto the cage during its final temperature excursion, and it was this grease residue which caused most of the high torque observed at the 871-hour inspection.

The ball pockets were heavily worn, with silver removed in substantial areas. Since the cage had originally been ball-riding and had gradually become partially land-riding, the contact areas in the pockets were large and nonuniform. One segment of the bore of the cage had contacted outer lands. Maximum depth of wear was approximately .002 inch.

Raceways showed bands indicating operation in different thermal regimes, but no apparent depth of wear. Balls showed a slightly mottled appearance, indicating small areas of surface distress which must have occurred at the end of the test. Rings were slightly discolored from having been subjected to moderately high temperature.

Figures 84, 85, 86, and 87 show components from Bearing S/N 6 under low magnification.

Figure 84 shows a section of the inner raceway. The ring was slightly discolored due to moderately high temperatures experienced by the bearing, but the ball path was burnished.

Figure 85 shows wear in the ball pockets. This wear is concentrated around the bore of the cage, but fore and aft in each pocket it extends radially outward. Dark grease residue remains in the pockets.

Figure 86 shows wear in the bore of the cage. The depth of wear is very close to the thickness of the silver plate (maximum .002 inch) sometimes penetrating to the S-Monel base. Two partially removed rivets also show.

Figure 87 shows wear on the OD of the cage. The depth of wear is very close to the thickness of the silver plate (maximum .002 inch), with streaks of silver showing through the areas of heaviest erosion. Two partially removed rivets are also shown.

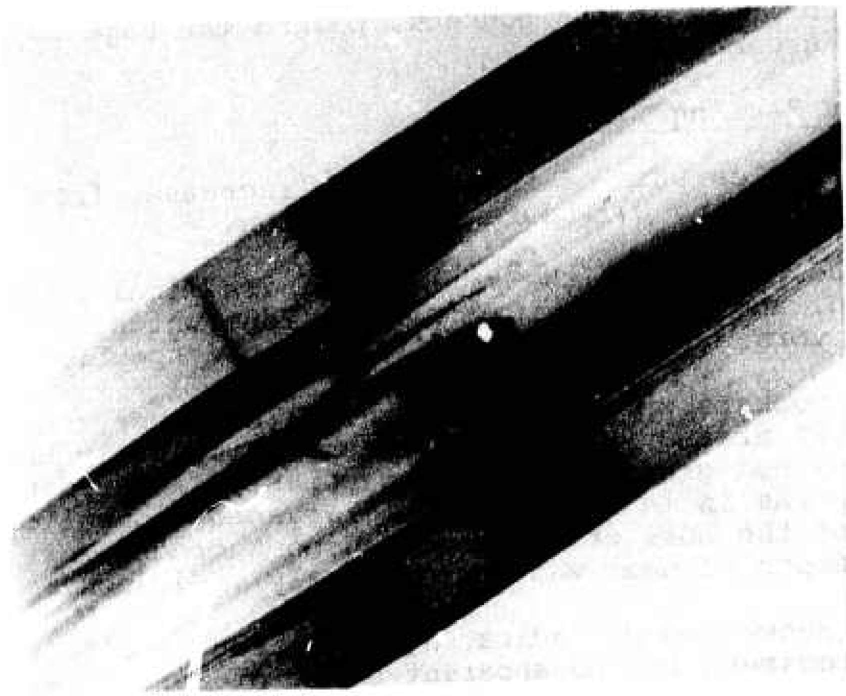

Figure 84. View of Inner Race, Bearing S/N 6.

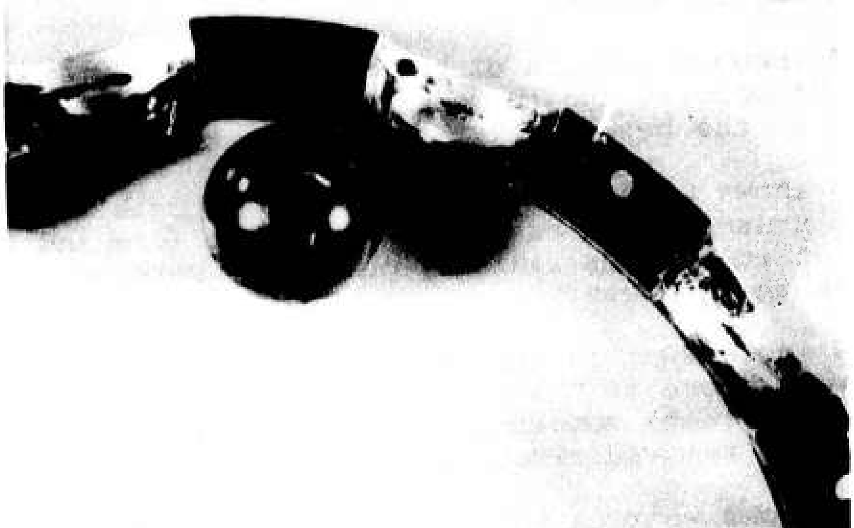

Figure 85. View of Cage Pockets and a Ball, Bearing S/N 6.

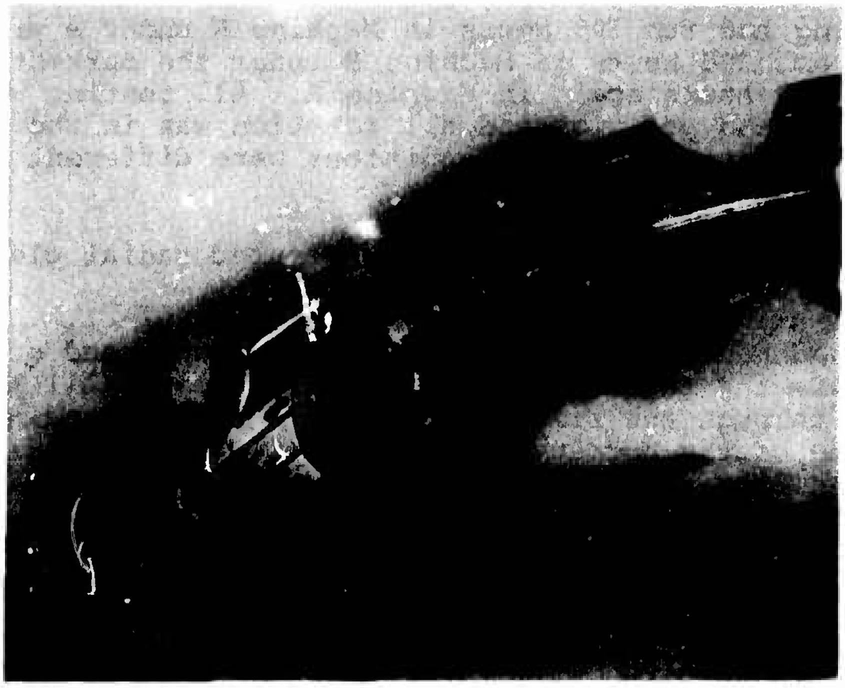

Figure 86. View of Cage Bore, Bearing S/N 6.

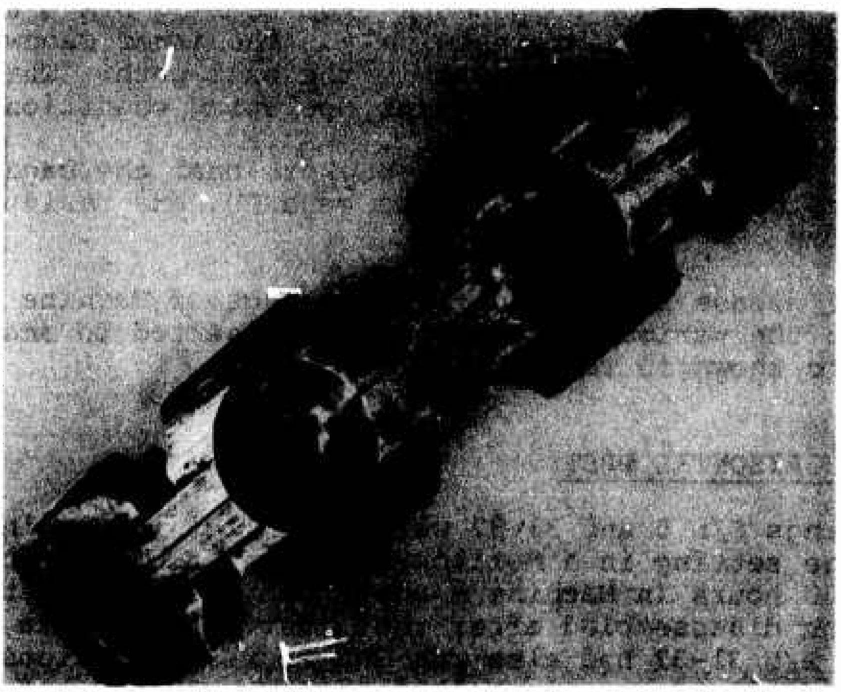

Figure 87. View of Cage O.D., Bearing S/N 6.

Drive-End Bearing S/N 5

This bearing had run 300 hours in Machine A under a previous test program, 300 hours in Machine B under the current program, and 129 hours again in Machine A. All testing was done in the drive-end position and all rotation was in the same direction; however, loading conditions were different between Machines A and B.

During the 129 hours in Machine A, measured radial clearance decreased from .0016 inch to .0015 inch.

The cage had made very light contact with inner and outer lands, polishing the silver in two small areas. All pockets were worn through the silver, 360°, in a narrow band near the bore. Some of the pockets indicated ball contact radially higher than others, showing that pocket wear was permitting cage location to change.

The outer race showed a dark band all the way around near the center of the raceway, and the inner race showed a corresponding band that was partially obliterated during subsequent operation. Otherwise, ball paths were cloudy and indicated the expected combination of thrust and radial loads, and misalignment. Balls were in good condition.

Figures 88 and 89 show components from Bearing S/N 5 at 3.2X and 2.4X magnification, respectively. The inner raceway in Figure 88 shows differentiation in the ball path. This bearing was subjected to two different operating conditions.

Figure 89 shows wear in the cage pockets near the cage bore. Other discolored areas, toward the cage OD, are residual grease films.

Samples of grease from inside the housings of Machine A were removed at the various inspection and subjected to analysis. Results are shown in Table VI.

300-HOUR HORIZONTAL TEST

Test bearings S/N 5 and 31-32 were installed in Machine B, with the machine setting in a horizontal orientation. Bearing S/N 5 had run 300 hours in Machine A under the previous test program and was not disassembled after that test. Components, except balls, of S/N 31-32 had also run 300 hours in previous testing. Bearings S/N 31-32 were installed in the outboard end of the spindle, and S/N 5 was run in the drive-end. Each housing was filled with 7-1/2 ounces of grease.

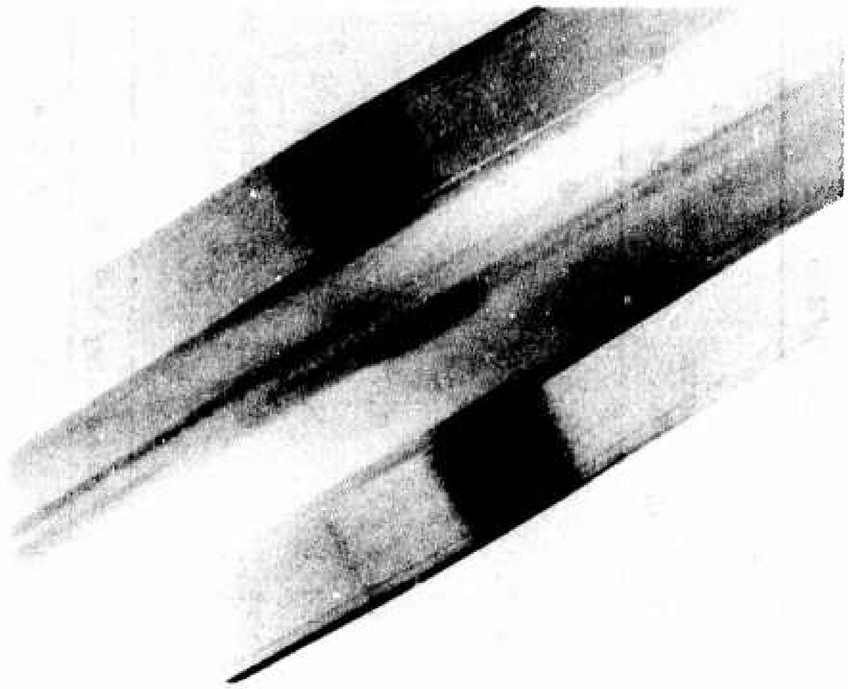

Figure 88. View of Inner Race, Bearing S/N 5.

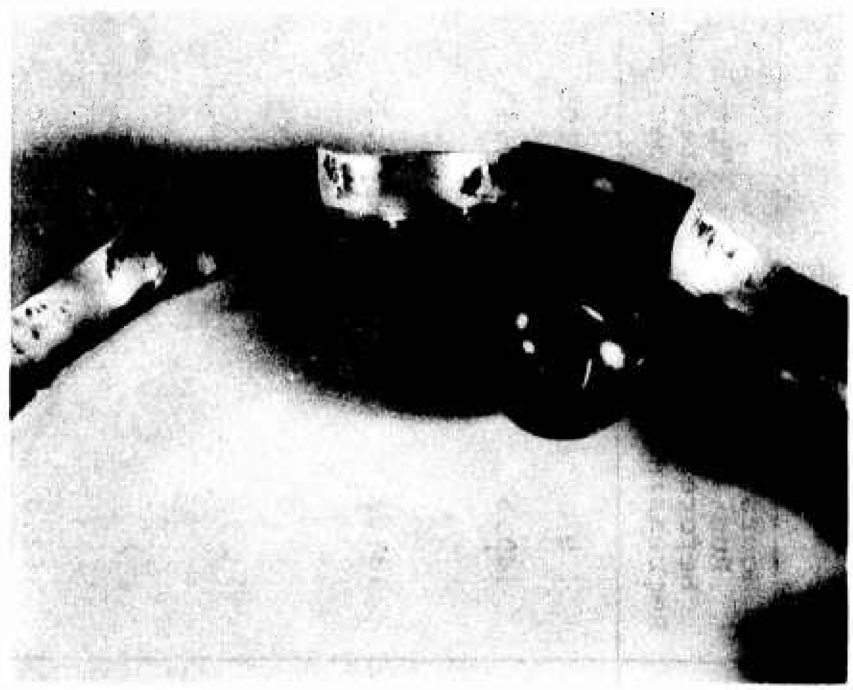

Figure 89. View of Cage Pockets and a Ball, Bearing S/N 5.

TABLE VI. GREASE SAMPLE ANALYSIS FROM TEST MACHINE A

Total Test Time (hours)	Bearing S/N 1			Bearing S/N 6		
	Hours Run Before Regreasing	Acid No.	Total Metallics (mg per gram)	Hours Run Before Regreasing	Acid No.	Total Metallics (mg per gram)
Prestart	0	0.52	0.00	0	0.52	0.00
300	300.0	1.03	0.15	300.0	1.16	0.25
497.2				197.2	1.29	0.21
637.8	337.8	1.16	0.20	140.6	0.77	0.34
745.9				108.1	1.03	0.41
871.0				125.1	Highly contaminated but sample too small for analysis.	
				Bearing S/N 5		
1000.0	362.0	0.90	0.43	129.0	0.90	0.51*

*Apparently non-homogeneous sample, since visual inspection of grease indicated better condition than several other samples.

The test ran 300 hours until scheduled shutdown without regreasing. For most of the test, two heat lamps were used on the drive-end housing; only one was used on the outboard housing.

At the conclusion of testing, both housings had at least 6 ounces of the original 7-1/2 ounces of grease remaining. Grease in the drive-end housing was little changed in appearance from the original condition; grease in the outboard housing was slightly discolored.

Both bearings appeared to be in good condition and were not immediately disassembled. Measured radial clearance of bearing S/N 5 had decreased from an original .0017 inch to .0016 inch; that of bearing S/N 31-32 had changed from an original .0017 inch to .0018 inch. Bearing S/N 5 was later run for 129 hours in Machine A, then disassembled. Description of components of bearing S/N 5 is covered under "1000-Hour Test".

When the outboard bearing, S/N 31-32, was disassembled, it was found that the bore and OD of the cage had contacted the lands of the rings, but the contact was very light with no measurable depth of wear. At bores of one-half of the cage, there were indications of removal of tiny pieces of material in chips. This chipping resulted in a rather uneven wear pattern in the ball pockets, as shown in Figure 90.

Races and balls were in good condition. Ball paths indicated light thrust in addition to misalignment and a light radial load.

Figure 90 shows a section of cage and a ball from bearing S/N 31-32. Pocket wear is concentrated near the cage bore. Small black spots at the bore in some pockets indicate chipping of material. Dark areas in the pockets toward the OD of the cage are films of residual grease.

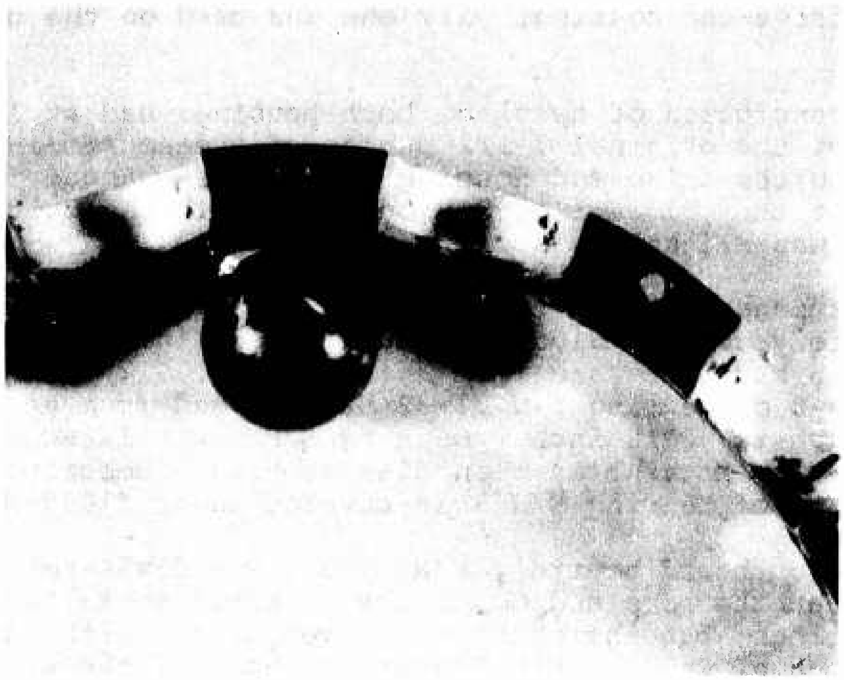

Figure 90. View of Cage Pockets and a Ball, Bearing S/N 31-32.